BOILER

ENGINE TESTS AND BOILER EFFICIENCIES

ENGINE TESTS AND BOILER EFFICIENCIES

BY

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TRANSLATED AND EDITED FROM THE THIRD EDITION
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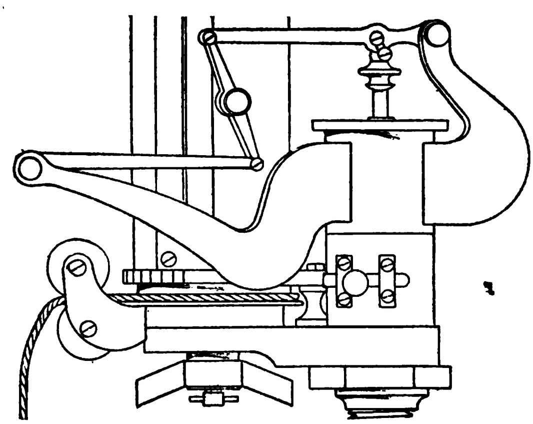

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AUTHOR'S PREFACE TO THE THIRD EDITION

IN this edition I have made considerable alterations and additions in almost every chapter of the book, especially in those parts which are concerned with the theory of indicators, the analysis of the working of their various parts, the description of new apparatus, such as indicators, with exterior springs for use with high pressure engines or when the steam has been superheated, explosion recorders for gas, petrol or alcohol engines, apparatus for reducing the scale of the reciprocating motion and the methods of setting it up, apparatus for verifying the flexibility of the springs of the indicators, etc. I have also made some additions to the chapter on brakes. In the second edition I had added a chapter on transmission dynamometers for use in measuring the work transmitted from the prime mover to the machine which it drives. This chapter, however, grew so large, especially that part of it which dealt with the theory of the dynamo-brake, that it appeared better to make this the subject of a separate book.

J. BUCHETTI.

PUBLISHER'S NOTE

IT has been thought that a translation of Mr. Buchetti's standard work "Guide pour l'essai des moteurs" should prove useful to English and American engineers, as it would enable them to compare the best continental practice with that in use in their own countries. With this end in view the measures and tables have in every instance been converted into English units. The formulæ also have been adapted into English measure. It is unnecessary for us to speak of the great value and striking originality of Mr. Buchetti's work.

We are indebted to the Hon. C. A. Parsons, F.R.S., for a brief chapter on testing steam turbines, a subject which is not touched upon by the author.

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CHAPTER I.

INDICATORS.

A BRIEF REVIEW OF THE VARIOUS INDICATORS INVENTED PRIOR TO THAT OF RICHARDS.

Watt's Indicator.

THIS apparatus (Figs. 1 and 2) consists of a small bronze cylinder of from 1·5 to 2 inches in diameter in which works a plunger, also of bronze, on the upper surface of which rests a spiral steel spring. The piston rod, passing through the spiral spring and a guide bracket, has a pencil holder attached to the end of it, which holder, by means of a spring, keeps a lead pencil pressed upon a wooden board covered with a sheet of white paper.

This board is free to slide to right or left in a frame, being pulled in one direction by means of a counter-weight, and in the other by means of a cord, the pull on which corresponds to the movement of the piston of the engine which is to be tested.

Whilst the cock which is fixed between the indicator cylinder and the cylinder of the steam engine is shut off and no steam passes, the pencil will trace a line *a–e* on the paper, corresponding to the atmospheric

pressure on either side of the piston. On opening the cock the plunger will rise during the admission of steam and compress the spring to an extent corresponding to the steam pressure behind it; whilst,

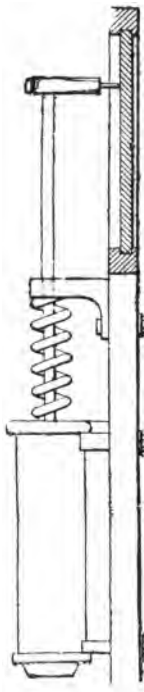


Fig. 2.

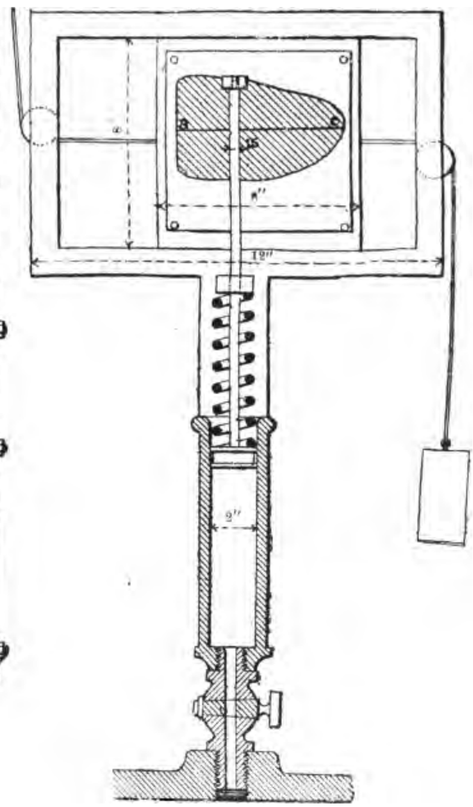


Fig. 1.

when the steam is exhausting, and as the sliding board moves back, the plunger will sink below the atmospheric line *a–e*, extending the spiral spring if the engine is a condensing one.

For each complete revolution of the engine, the pencil traces a diagram, of which the *abscissae* are

proportional to the crank throw—or travel of the piston—and the *ordinates* to the steam pressure throughout the forward and return stroke of the piston. This diagram represents the power developed during one revolution of the engine on one side of the piston.

MacNaught and Hopkinson Indicators.

In the place of Watt's sliding board and counterweight, MacNaught used a drum with a rotary action (Fig. 3) and an interior spring, the cord in this case passing over a grooved pulley at the base of the drum.

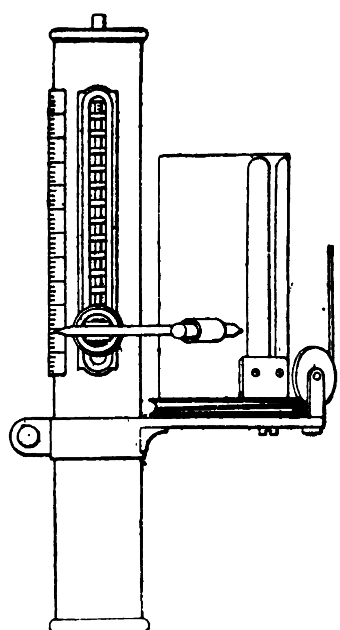


Fig. 3.

The pencil holder is attached to an arm connected to the rod of the plunger, and a spring keeps the lead constantly pressed upon a paper fixed round the drum. In this case it is difficult to regulate the pressure of the pencil or to lift it from the paper. To get over this difficulty, Hopkinson modified MacNaught's pencil holder (Fig. 4). Here the pencil d is carried by a rod, which is free to revolve round the spindle c and carries at its other extremity another rod b. By means of a lever a the

pencil may be lifted off the paper at will, whilst the indicator is at work. Adjustment is effected by means of a thumbscrew.

The Indicators above mentioned, in which the pencil is rigidly attached to the plunger, have long since been abandoned, for they are useless for engines of even moderate pressure, owing to the irregularity of the diagrams produced.

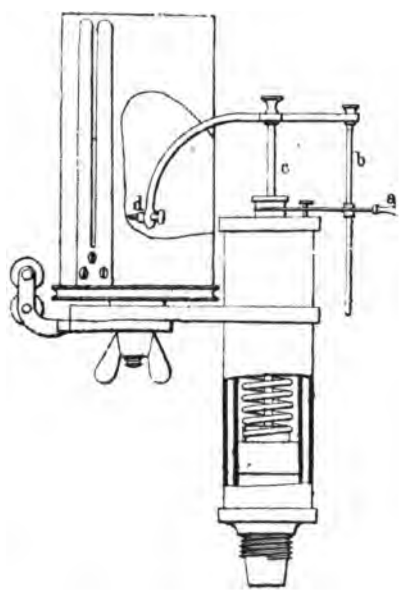


FIG. 4.

To get over this difficulty, Garnier (Figs. 5 and 6) fixes two collars, *i–i*, to the rod of the plunger, allowing an amount of play *y* which represents the throw of the plunger between them and the bracket *b*. The bracket *b* is moved forward by means of the bevel wheels *M N*.

Assume the pencil to be lifted off the drum, the spring to be compressed and the drum in motion—now press the pencil down on to the drum, and it will trace a series of ovals if the movement is steady or of horizontal lines if sudden and intermittent. But, in actually pressing down the pencil, the spring is slackened and its tension rapidly adjusts itself to the pressure of the steam, the plunger moves to the extent of the space *y* and the pencil traces at each

revolution of the engine a pattern which is the full extent of the first elements of the diagram.

FIG. 5.

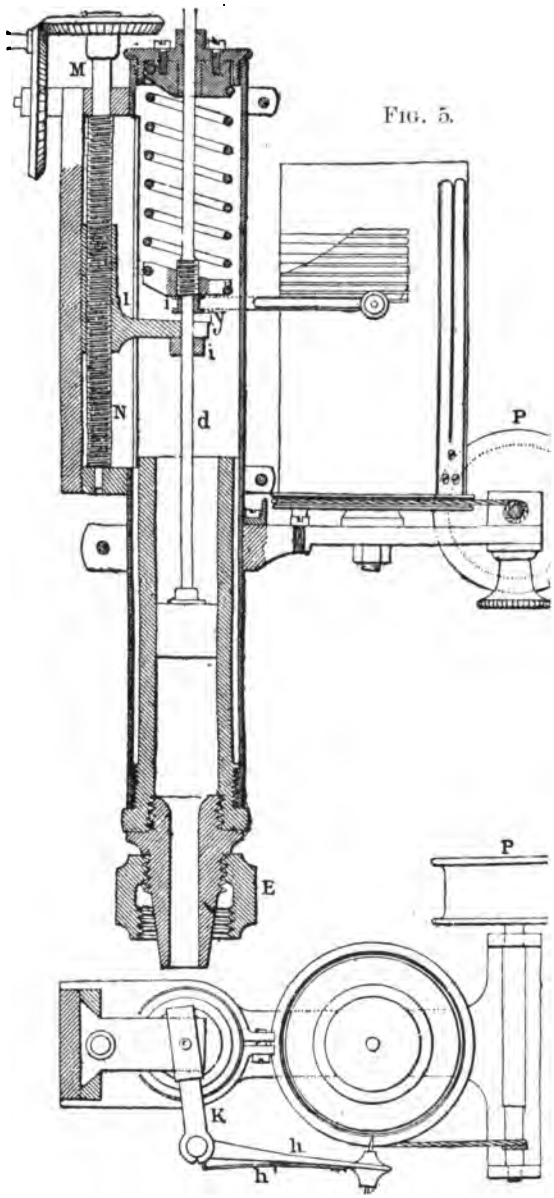


FIG. 6.

Continuing this, succeeding elements are traced of which the whole constitutes *the diagram of the mean pattern* produced by a definite number of revolutions.

This indicator is now out of date.

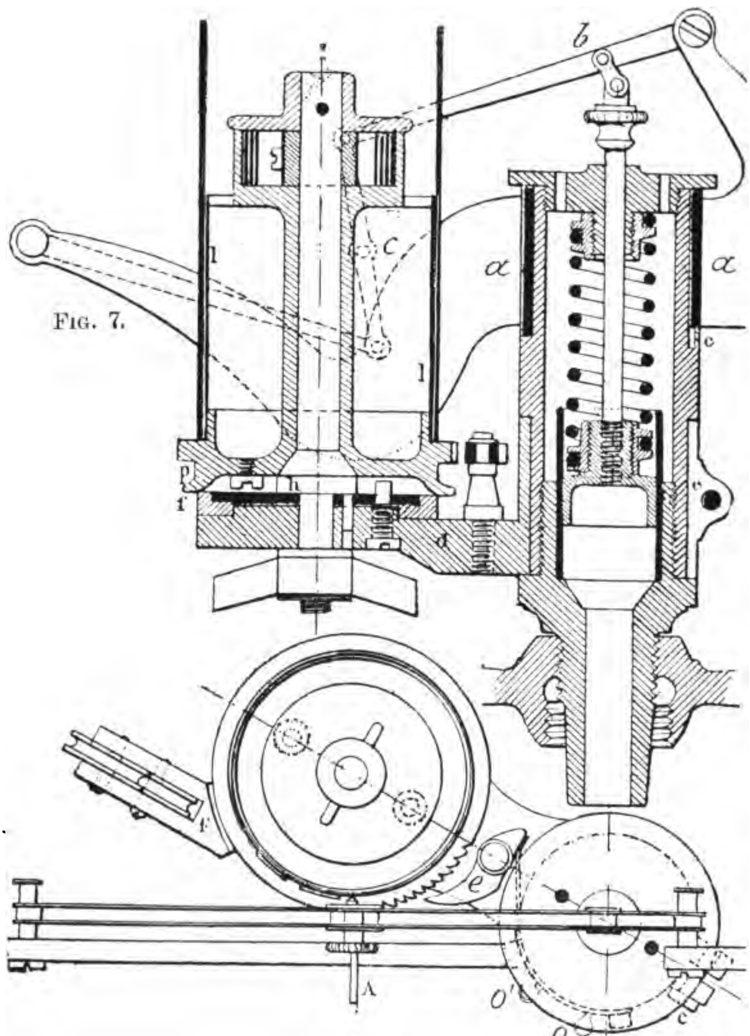


FIG. 7.

FIG. 8.

INDICATORS

Richards' Indicator, manufactured by Elliott Bros. (Figs. 7 and 8).

This indicator is an important advance upon those already described, and is applicable to engines of high steam pressure and high speed.

Here the travel of the plunger is reduced to about three-quarters of an inch, whilst a lever, whose arms are in the ratio of about 1 to 4, permits the tracing of a diagram with sufficiently long ordinates. A pencil *A* attached to a parallelogram traces within the limits required of it a practically straight line. The lightness of the levers carrying the pencil reduces their inertia, which is very small compared with that of the plunger, and hence, notwithstanding the increase of travel of the pencil, the consequent natural tendency to exaggerate every oscillation is very slight.

The collar *a–a* which carries the parallelogram, and the link *b*, are each free to move round the axis, and hence the pencil can be pressed or lifted from the drum, even while the plunger is at work. This movement is limited by a stop *c*.

Holes drilled in the cover of the cylinder admit air to the upper side of the plunger and allow of its escape as the plunger rises.

The washer *f* carries two pulleys, seen in the plan, which act as guides to the cord, from whatever direction the pull comes upon it. A movable drum which carries the paper for the cylinder fits over and is attached by a stud to the cylinder, which is fixed to the pulley *p* and the barrel containing the counter spring.

The drum may be fitted with *Darke's* ratchet

adjustment, by which means its movement is arrested when a complete movement to and fro has taken place, so that the paper may be changed without disconnecting the cord.

This ratchet is thrown in or out of gear by the movement of a button from O to O′.

The Richards' Indicator is the model followed by all subsequent inventors.

Darke's Indicator, manufactured by Elliott Bros. (Figs. 9–12).

This little Indicator is designed for engines of high steam pressure and high speed.

The diameter of the plunger is here reduced to half an inch, and its throw to about two or three eighths of an inch; the diagrams are at most one and a half inches deep by three and a half inches long.

The hollow plunger rod has at its upper end two flanges, which hold a socket between them by means of two screw pins. The enlarging lever *b* passes, an easy fit, through this socket. One end of this lever is attached to a bracket fixed on the collar *e*, whilst to the other end, which is flattened out, is attached a slide *c* carrying a pencil which works in a straight groove cut in the plate *d* which is also attached to the collar *e*.

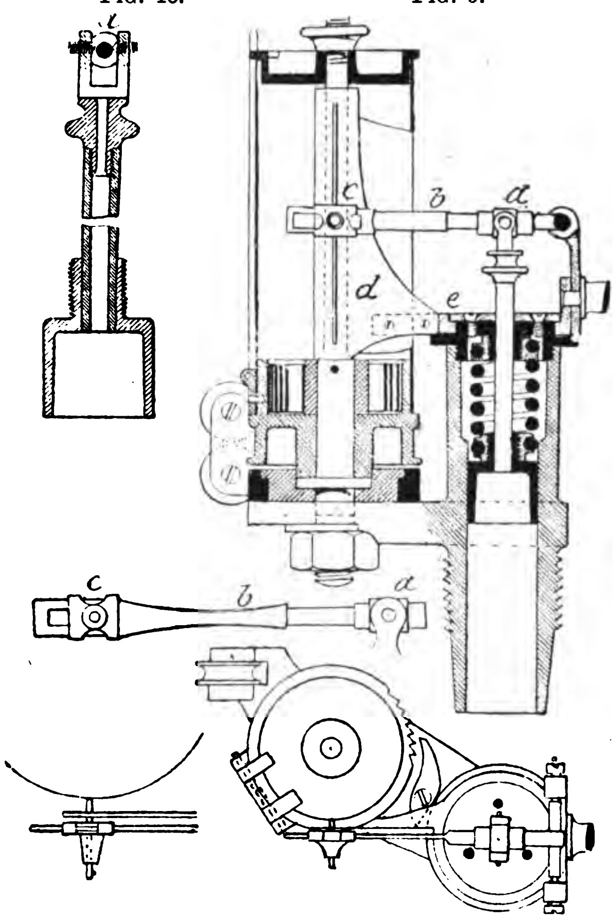
The pencil is moved on to or away from the paper by means of a small handle which turns the collar *e* and lifts the lever *b* and guide plate *d*. The drum, provided with the *Darke* ratchet adjustment, carries the paper.

After taking each diagram, the drum must be stopped and a new piece of paper, bent to the diameter

of the drum, slipped over it and kept in position by means of two hinged plates provided with teeth on their upper surfaces, on to which each end of the paper is folded back.

Nearly all makers of Indicators manufacture two

Fig. 10. Fig. 9.



Figs. 11, 12.

types, the one for low, the other for high speeds and steam pressure.

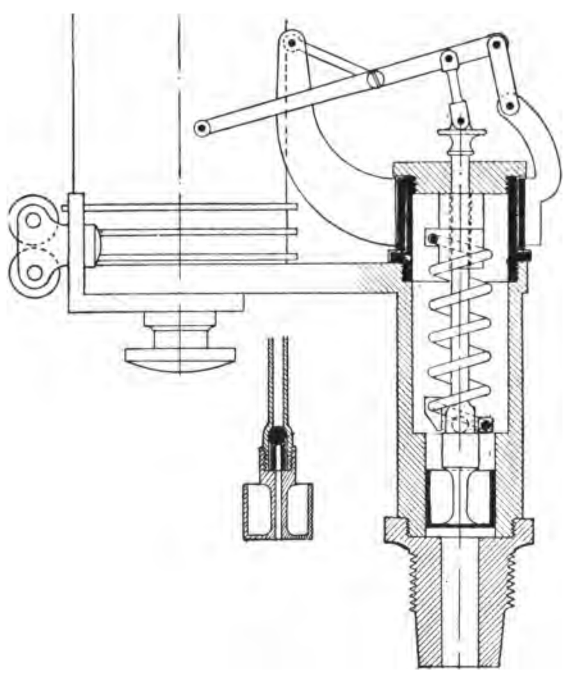
Thompson's Indicator (*American*).

This Indicator only differs from that of *Richards'*, by using a parallelogram which is practically that used by Evans, except that it has a short connecting rod. Evans' parallelogram, which is simpler than that of Richards, has less inertia, and so the irregular movement of the pencil is less.

Chapter IV. will describe the conditions to which it is most applicable.

P. Garnier's Indicator (Figs. 13, 14).

This little pattern is constructed for high speed



Figs. 13, 14.

engines with high steam pressure, and is fitted with Evans' parallelogram. The upper surface of the

plunger, which is made of steel, is cup-shaped, and is filled with oil which is at each movement to and fro thrown over the inner walls of the cylinder. The lower end of the spring fits into a cap with a ball in the centre. This ball fits in between the plunger and the hollow rod screwed to it. Thus the tension of the spring can be altered by simply unscrewing the piston, when it has been taken out of the cylinder, without taking the rod off and without using a spanner. The drum is provided with two grooves, the upper carrying a cord which can work a second indicator.

Care has to be taken with the indicator previously dealt with not to bend the parallelogram, as this would alter the spring, for the cover of the cylinder which has been unscrewed and the socket carrying the parallelogram are both loose in the hand. In this indicator, however, the upper part of the cylinder to which the cover is screwed and to which the socket carrying the parallelogram is fixed, can be unscrewed and entirely disconnected from the lower part.

The Martin-Garnier Indicator (Figs. 15–17).

This Indicator is distinguished from the preceding ones in the first place by the fact that the drum is worked by means of a small wooden pulley with the cord wound on it and a pinion and helical toothed wheel. The spindle of this pulley has at the other end a return action spring attached to a fixed band. The pulley can be changed to suit the requirements of the engine under test, and the apparatus suitably reduces the effect of the throw of the crank shaft of the engine, and has the additional advantage of stiffening

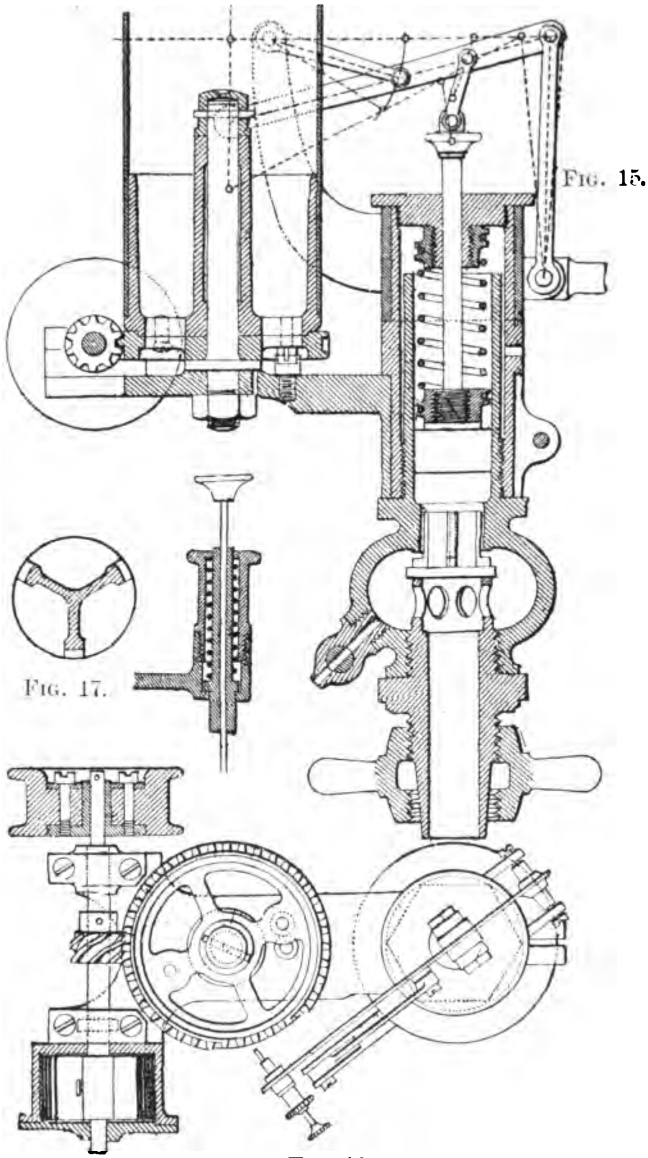


FIG. 15.

FIG. 17.

FIG. 16.

the drum in the case of high speeds, and thus increasing the correctness of the length of the diagram; on which point we will touch later.

In the second place, this indicator is distinguished by having a valve at the base of the cylinder. The inventor claims that this valve is lifted by any sudden steam pressure, and that it then checks the flow of steam into the cylinder; the plunger is therefore saved from any sudden inflow of the steam to the cylinder, and any tendency to irregularity in its movement is reduced to a minimum. Nevertheless the beneficial effects of this valve are questionable, and the need for it would only appear to exist in the case of tests of engines of very high speed.

P. Garnier's Magazine Drum, with Steadying Gear (Figs. 18–21).

This steadying gear does away with irregular movement of the drum at high speeds as well as those drawbacks attendant upon the vibrations and elasticity of the cord (see Chapter IV). It is eminently suitable, therefore, for high speed engines. The ratchet *a* (Figs. 18–19), which has a to and fro motion transmitted to it by the engine, gears into a pinion *b*. This pinion *b* with its sheave *c* is loose on its spindle, and the rack can be moved backward and forward through the guides through which it passes.

The sheave *c* (Fig. 20) forms, with the sheave *d* fixed on its spindle, a clutch, which is thrown in or out of gear by the lever *e*.

The fixed sheave *d* (Fig. 21) mounted on a squared portion of the spindle, when thrown into gear with *c*, sets in motion the pinion *f* with helical teeth, geared to a similar pinion fixed to the spindle of the paper drum *k*. The barrel *g* contains a return action spring, giving

the return motion to both the rack a and the drum k.

Inside the paper drum k is a spool of paper which is

Fig. 20. Fig. 18.

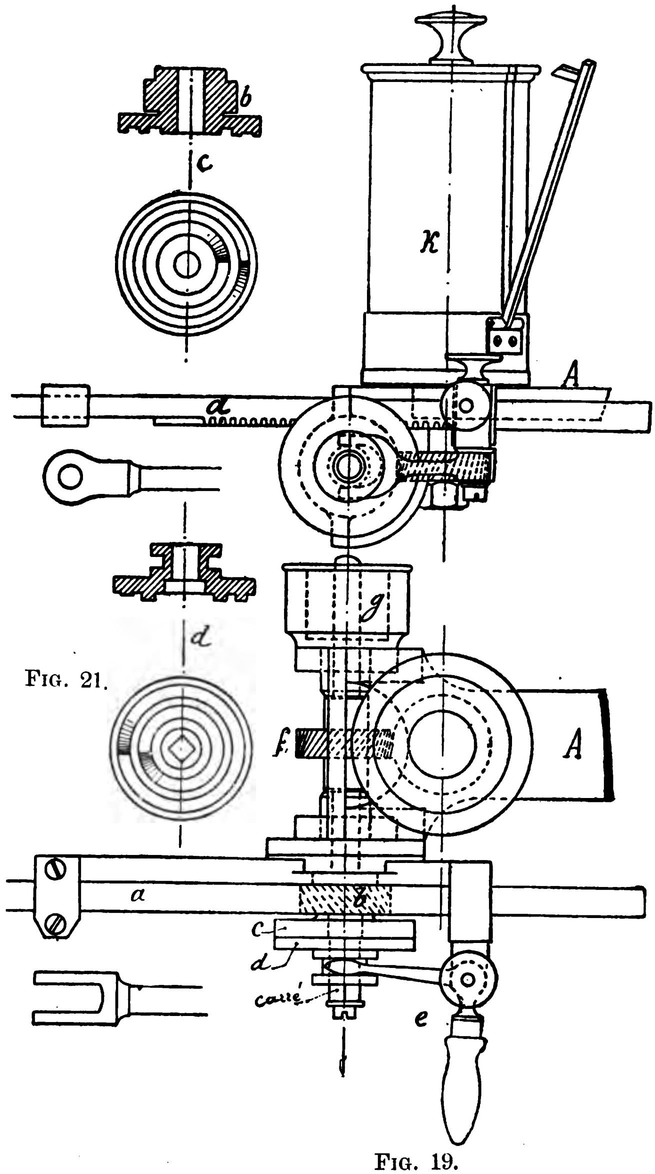


Fig. 21.

Fig. 19.

reeled off on to the drum as required, and which is held in position by closing down the lever blades. As each diagram is traced, the apparatus is thrown out of gear by the lever *e* and the paper torn off along the blade; the blade is then lifted, a new length of paper drawn out and fixed round the drum, the clutch thrown into gear again and a new diagram taken.

Schaeffer and Budemberg's Indicator (Fig. 22).

In this Indicator, the hollow piston rod, which is of steel, works in a guide at the head of the cylinder, and the connecting rod which connects the piston rod to the multiplying lever is also made of steel, and has a ball and socket joint. A cord actuates the paper drum in the usual way.

The cap of the barrel containing the return action spiral spring for the drum is loose on the spindle, and is fixed to the inner end of the spring.

To stretch the spring to the required extent, all that has to be done is to slacken the upper guard

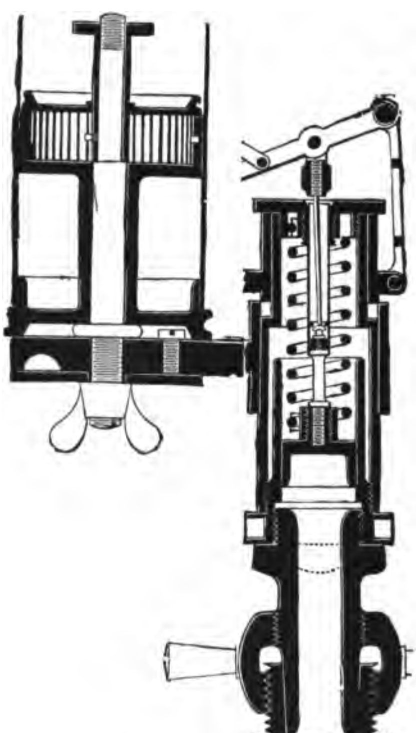


FIG. 22.

nut, give a turn to the cap of the barrel and tighten down the nut again.

Rosenkranz's Indicator (Fig. 23).

Here an interior cylinder *a* steam jacketed to a depth equal to the whole travel of the plunger keeps the walls of the cylinder throughout the stroke at an even temperature, and prevents any unequal expansion between the plunger and the cylinder. In order that the expansion should have free play throughout the material of the inner cylinder, there is a clearance between its upper portion *b* and the outer cylinder. With this arrangement, the plunger never jams in the cylinder, even when the latter has not been heated beforehand.

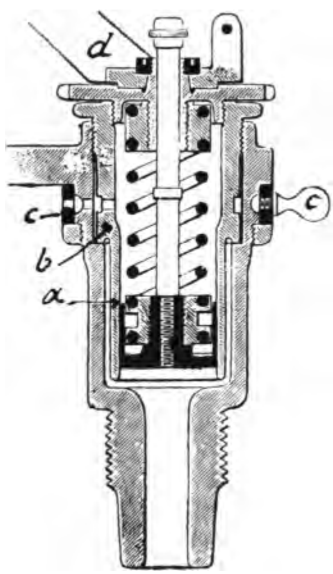


FIG. 23.

The arm *d* carries the guide rod of the parallelogram. The nut *c* is used to draw away any water caused by condensation in the cylinder.

Crosby's Indicator (Figs. 24, 25).

The spring (Fig. 26) is in this case a double spiral, with a ball fitting on to it at the base. The socket which prolongs the piston is split to receive the spring, and the ball is held firmly between the screw placed inside the socket (Fig. 24) and the

piston rod proper, which is screwed into the socket after the spring is inserted.

Several holes drilled in the cylinder, above the

Fig. 24.

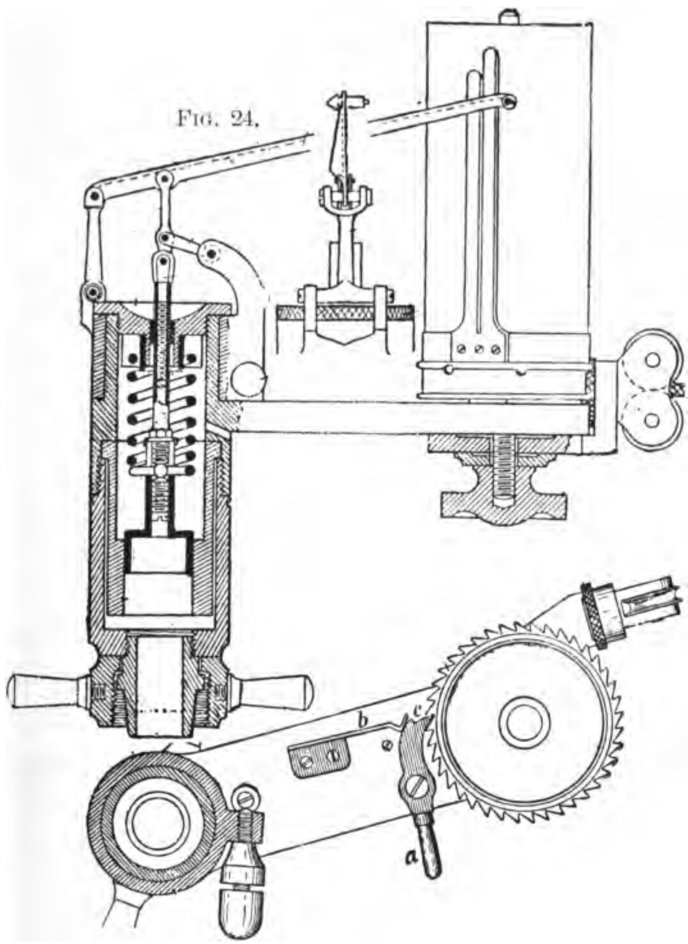


Fig. 25.

plunger, prevent the air cushioning, and they also allow any water which may have formed by condensation above the piston to escape. The cylinder is

steam jacketed, as in the Indicator last described, in order to secure equal expansion.

The parallelogram is of special design, and its functions will be found described in Chapter IV.

FIG. 26.

In this Indicator the lever guide is smaller, and nearer the axis than that of Evans', and its inertia is consequently less. The long lever arm has a web which serves to stiffen it.

The drum is set in motion in the usual manner; the spring *b* is first lifted, then the handle *a* is pressed down and the elbow of the spring fitting into the niche *c* holds the ratchet out of gear. The object of this ratchet is, as has already been explained, to stop the drum without disconnecting the cord which works it.

Elliott Brothers' External Spring Indicator (Figs. 27, 28).

For use with engines driven by superheated steam of very high temperature, Messrs. Elliott Brothers have constructed an Indicator in which the spring is placed outside the cylinder in order that its hardness, and consequently its elasticity, shall not be affected.

The spring is of special design, shaped like tongs, the two ends being bent round and fitting into sockets. The spring can be taken out and replaced without taking the apparatus to pieces.

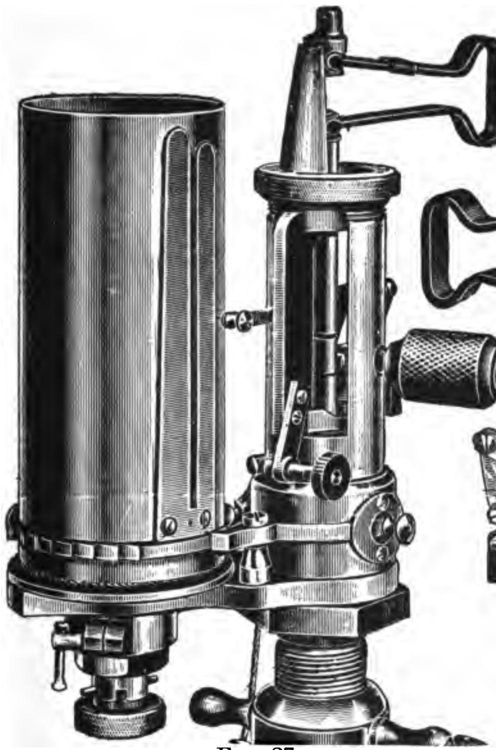


FIG. 27.

FIG. 28.

Rosenkranz's External Spring Indicator
(Figs. 29, 30).

This Indicator, as the last described, is designed for steam engines working at very high temperatures, engines using superheated steam, gas engines, etc.

The connecting rod *K* carries above it a stirrup *B* upon which fits the lower sheave *G* of a double spiral spring. The turned boss *M*, *R* fits into the upper sheave *G* 1. The hollow cap *A* is fitted with a milled and screwed cap *N*, which presses on *R* and holds it in place.

On the same piece as the hollow boss is a smaller

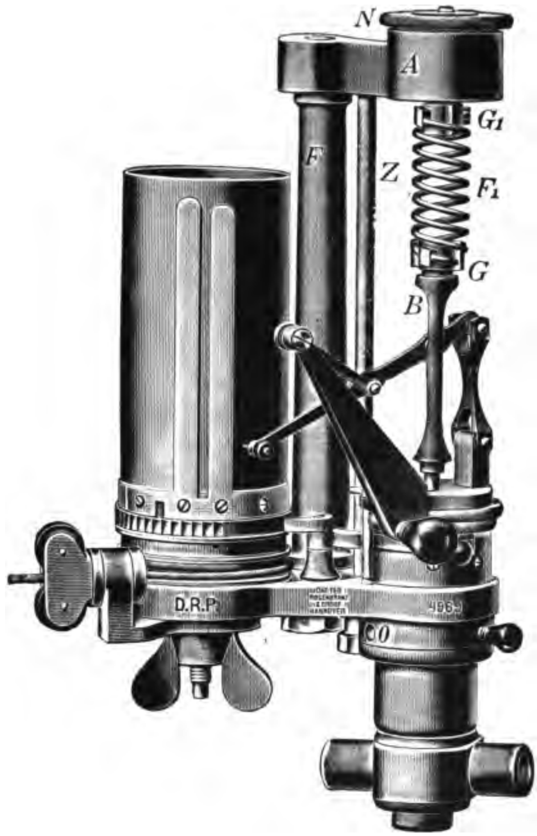


FIG. 29.

FIG. 30.

one which fits into the hollow steel column *F*, which is rigidly fitted to the base plate of the whole apparatus, the rigidity being increased by a tie rod *Z* also attached to the base plate. For the rest, the Indicator resembles those already described. The weight of the stirrup *B* is of no importance. To change the spring, the screwed cap *N* must be removed after the screws which hold the cap of the cylinder have been taken out. The plunger can then be withdrawn from the cylinder, and the head *R* from

the hollow boss *A*. There is no difficulty in this, as the external parts keep cool, and as the lever arms are not interfered with, the parallelogram need run no risk of injury.

Kenyon's Indicator (Fig. 31).

Mr. Bourdon, with his well-known hollow spring

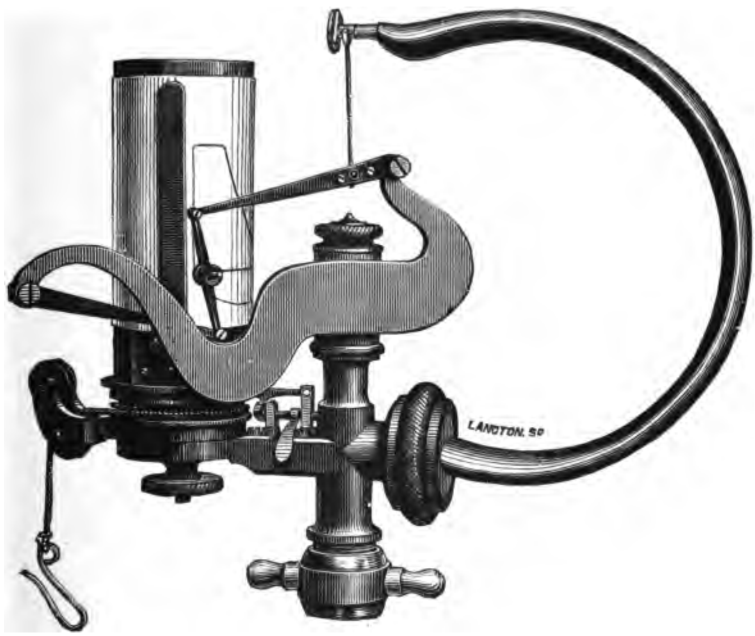


FIG. 31.

device, constructed an Indicator of which there is a specimen at the Conservatoire.

Kenyon, taking up the idea, applied Bourdon's hollow spring to the Richards' Indicator. The flexibility of the spring is varied to suit the pressure of the steam in the engine under test. This apparatus obviates all difficulties arising from friction between the plunger and the walls of the cylinder, and all discharges of hot

water which interfere with the operator and so often wet the paper and cause it to tear.

But as the values of the angles of deflection can only be plotted under steam, compressed air or water pressure, the apparatus has been discarded.

Prussman's Indicator, manufactured by Schaeffer and Budenberg (Fig. 32).

The diagram traced by all the Indicators above described gives the varying pressure on one side only of the piston of the engine, and it is often taken for granted that the pressure is the same on the other side as well.

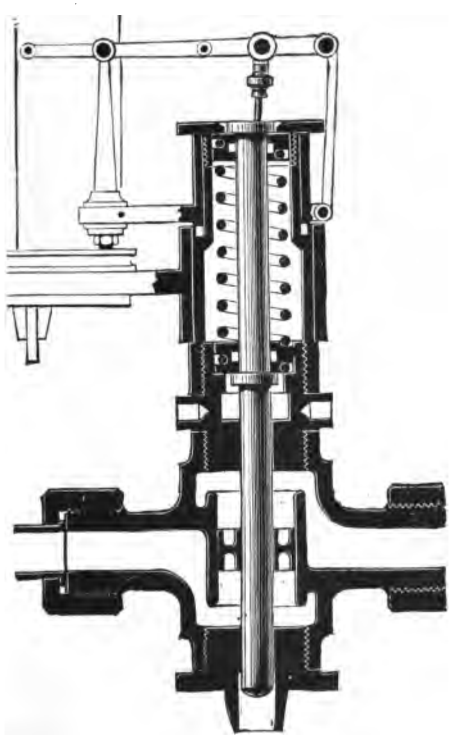


Fig. 32.

This is not true, for the slanting stroke of the connecting rod causes a difference in the distribution and speed of the piston in its backward course.

In order to obtain the total work done by the steam on the two faces of the piston, it is necessary to take indicator diagrams on both sides of the cylinder. If

we arrange the indicator diagrams as shown in Fig. 33, and subtract from the ordinate of the curves showing the steam pressure for the forward strokes on each face the simultaneous back pressure on the other face, we get the shaded diagram in Fig. 33. For example, b' equals b, and c' equals c. This shaded diagram may be regarded as the true indicator diagram.

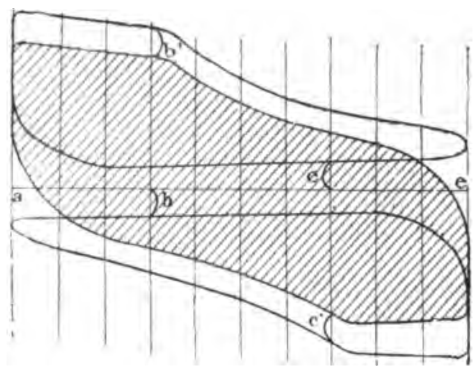


FIG. 33.

The differential Indicator actually gives this true diagram. One side of the plunger of the Indicator is connected to one end of the engine cylinder, and the other side to the other end. The plunger rises or falls with each stroke of the engine in proportion to the difference of pressure in the cylinder.

It will be seen from the drawing that the spring is always in compression, and therefore that the diagram given is that shown (Fig. 33), which is a combination of the true diagrams—back and front—and represents the work given out at each complete revolution of the engine.

The ordinates within the curves show the true varying pressure upon the piston of the engine.

If the Indicator is connected up separately, first to one side of the piston of the engine and then to the other, the two separate diagrams (Fig. 33) are obtained.

This Indicator can be altered from a differential to a simple one by stopping the left-hand steam inlet, and by replacing the bottom plug by a pipe and stopcock, but the diagram now is not so tall as an ordinary one, as the lever carrying the pencil is already on the base horizontal line at atmospheric pressure.

In our opinion, the true diagram is not as useful as an ordinary one, for one cannot judge of the distribution of steam in the cylinder from it, since the lines indicating admission, cut off, expansion and compression, which will be studied later on, are altered in shape by this fusion. The fact that the diagram shows the work expended on both sides of the piston, for each complete revolution of the crank is however of small importance, for if the engine runs smoothly and without change of load ordinary diagrams taken one immediately after the other will suffice. If on the other hand the engine is not running under steady load, it is necessary, even with the double indicator, to take several diagrams in order to arrive at the mean. When acting as a double indicator, the diagrams do not take atmospheric pressure into account. This advantage is really a slight one, owing to the very trifling changes in this pressure, and the stronger the spring, the smaller will be the effect of these changes.

CHAPTER II

INDICATORS WITH CONTINUOUS ACTION

CONTINUOUS Indicators—that is to say, indicators which give a continuous record—have not been adopted as practical.

They are more complicated in their construction and more costly than those we have described. Moreover, it is but seldom that a continuous record is really of advantage.

So long as we can take a simple diagram, once an hour, or even oftener, this is usually sufficient for ascertaining the output of an engine.

The simple diagram suffices to show the engine builder how to adjust cut-off and expansion—it is all that is required for an ordinary trial; nevertheless it will be well to describe some of the very ingenious Indicators that have been designed for taking continuous diagrams.

Guinotte's Indicator (Figs. 34–35).

This is an improved form of Clair's continuous Indicator, the improvement consisting in replacing Clair's direct acting pencil arm by that of Richard's.

The paper rolled on the drum *A*, which is fitted with a check *B*, passes to the drum *C*, which, by an

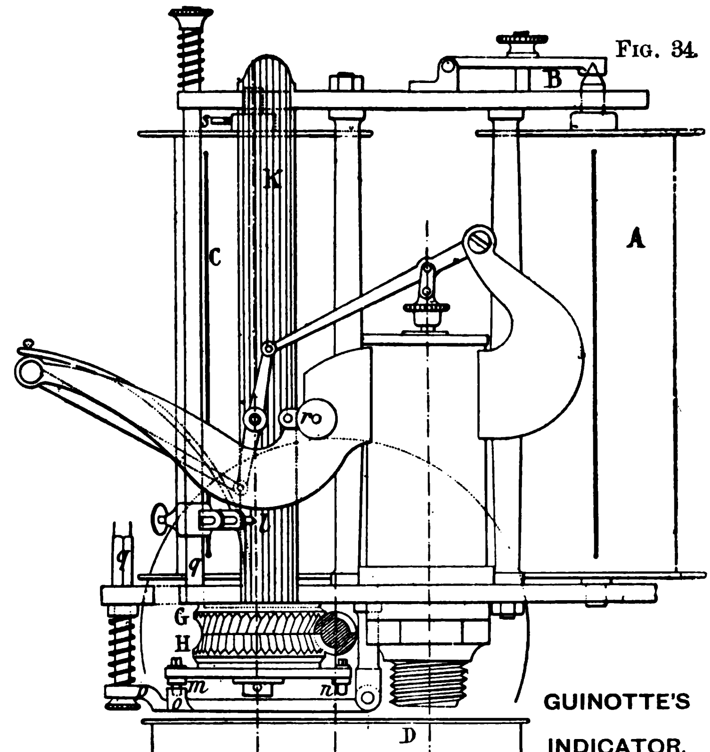


Fig. 34.

GUINOTTE'S INDICATOR.

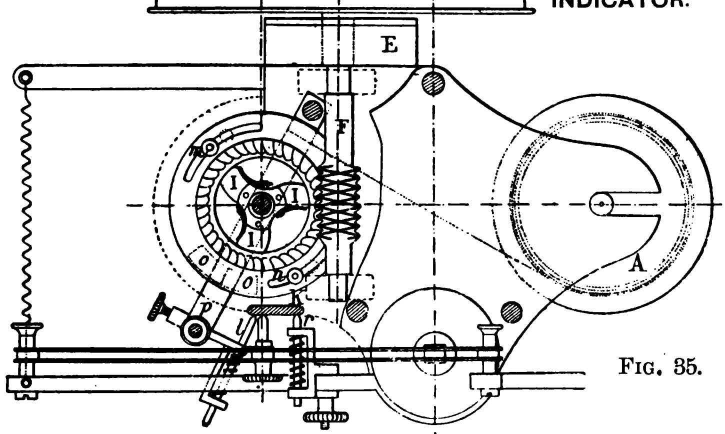


Fig. 35.

ingenious arrangement due to Clair, receives a steady rotary movement, proportional to the speed of the

engine under test. The pulley *D* is kept in motion by a cord or belt driven by the engine whilst the return action is given by the spring *E*. The spindle *E* moves, therefore, alternately from right to left. In order to convert this alternating movement into a continuous one at the drum *C*, the spindle *F* has a right and left screw thread cut in it, of which the right-hand thread is in gear with the bevel wheel *G* and the left with the bevel wheel *H*.

The drum *C* is given a continuous motion, each wheel driving it in turn by means of three friction clutches *I*, *I*, *I*, inside it, which by means of springs press on the inner periphery of the wheels only when these revolve in one direction.

Fig. 35 shows the upper bevel wheel removed, and shows the friction clutches inside the lower one.

In the early designs, the curves were traced on the drum *C*, and some inconvenience resulted owing to its increasing diameter as more and more paper was wound on to it. To get over this difficulty the paper was passed over a fixed blade *K*. As the drum *C* increased in diameter the diagrams lengthened, and Fig. 36 shows that given at each revolution, indicated by

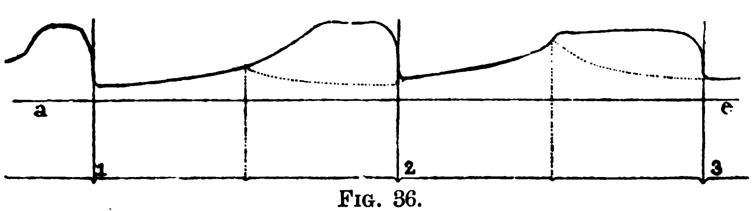


FIG. 36.

dotted lines 1, 2, 3, traced by the pencil *l*. This pencil *l* is worked by means of two bolts *m*, *n*, clamped into a flange of the wheel *H* (Fig. 34), and so placed

that at each end of the stroke of the piston each bolt alternately strikes one of the bevelled sides (o o) of the lever P, and sharply presses down the spindle q, which carries the pencil l, which pencil is during the rest of the time kept off the paper by means of a spring. In the diagram shown, Fig. 36, the compression curve follows the pressure curve; if we fold the one back upon the other we obtain the ordinary self-contained diagram.

The pencil r traces the line a–c at the same time that the curves are being traced, but we can trace this line after taking the diagrams, by loosening the screw bolt S which attaches the drum C to its spindle, releasing the check B and winding the paper back on to drum A by hand.

Recording Indicators.

Ashton and Storey's Indicator (Fig. 37).

In this Indicator each end of the cylinder A is connected to a different end of the cylinder of the engine under test by means of cocks fitted with blow-off valves.

The plunger A, the rod of which carries loose upon it the little wheel a, attached in its turn to the cylindrical pinion b, moves up and down in proportion to the difference of steam pressure on either side of it. This pressure is balanced by a spring C, the flexibility of which is proportional to the pressure for which it is used; a series of springs of different strength are therefore necessary, as in those indicators already described.

Each revolution of the engine is also transmitted

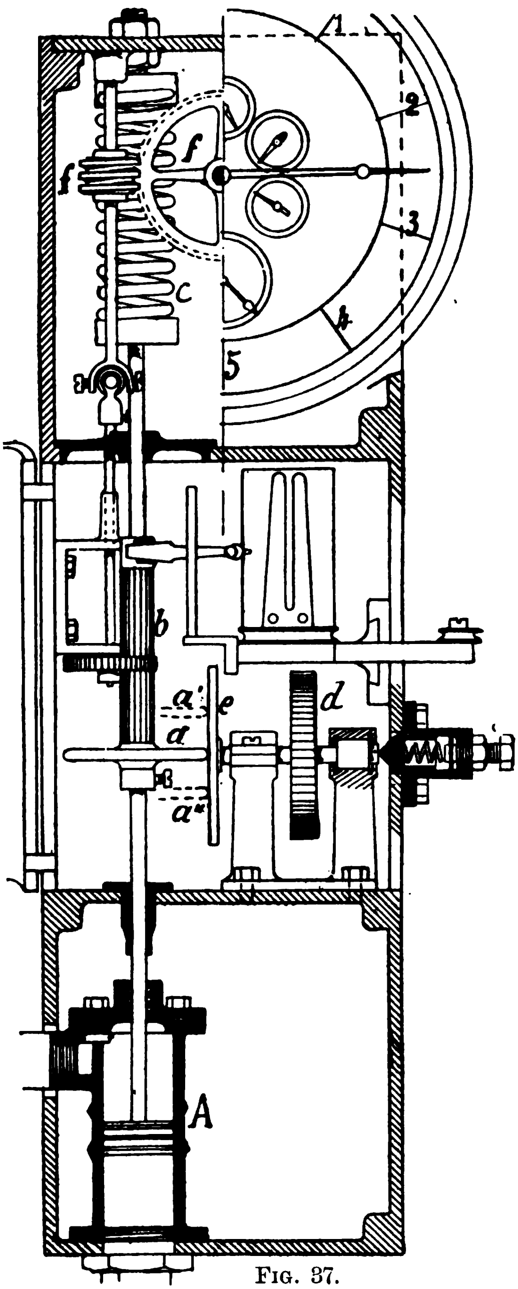


FIG. 37.

(by means not shown in the diagram) through the toothed wheel or pulley *d* to the disc *e*.

So long as the pressures on each side of the piston are equal there is no movement of the plunger *A*; the wheel *a* bears against the centre of the disc *e*, and takes no rotary movement from it, but as soon as the pressure varies, the wheel *a* moves to *a′* or *a″*, and revolves at a speed proportional to its distance from the centre of *e*.

The number of revolutions

made by a is proportional to the work of the engine and is recorded by the wheels $f\,f$ and read off on the dials.

Let us assume that the circumference of the wheel d is equal to the travel of the piston of the engine and that an effective pressure of 70 lbs. per circular inch causes the wheel a to rise to the position a'—i.e. half-way between the centre and the periphery of the wheel e; then for every foot covered by the engine piston, the little wheel will travel 0·50 of a foot.

If now, actuated by the clockwork, the pointer registers $\frac{1}{25}$th of the circumference of the little wheel a it covers $\frac{0{\cdot}50}{25}$ of a foot$=$0·24 of an inch, which represents 70 foot-pounds of work done by the engine for every circular inch on the face of the piston.

In these indicators each division of the dial represents 100 foot pounds; that is to say, 1,000 foot pounds for every complete revolution of the pointer per circular inch on the face of the engine piston.

Given therefore that

n' is the figure on the dial before commencing the test,

n is the figure on the dial after the test is complete,

D the diameter of the engine piston in inches,

m the duration of the test in minutes,

we have $100(n-n')\,D^2 =$ work given out in foot-pounds and therefore the average horse power (H) is given by:

$$H = \frac{100\,(n-n')\,D^2}{550 \times 60 \times m} = \frac{n-n'}{330\,m} D^2$$

and for a day's work of 10 hours, where $m = 600$—

$$H = \frac{n-n'}{330 \times 600} D^2 = K\,(n-n')$$

if $\dfrac{D^2}{198000} = K$, the constant for any given indicator.

These formulæ are true so long as the circumference of d is equal to the travel of the engine piston. If, on the other hand, c be the circumference of d and C the travel of the engine piston, the constant becomes $K\dfrac{C}{c}$.

To obtain absolute accuracy the rod of the indicator plunger A should extend on both sides.

In the case of two cylinder, or compound, engines, two separate indicators are used, or the connecting steam pipe is so arranged that steam from the high or low pressure cylinder can be admitted at will.

In the case of steady work being done by the engine, the work done in each cylinder may first be found separately, and their ratio calculated.

Let a = the work of the high pressure cylinder,

b = the work of the low pressure cylinder.

Then $a + b$ = the total work.

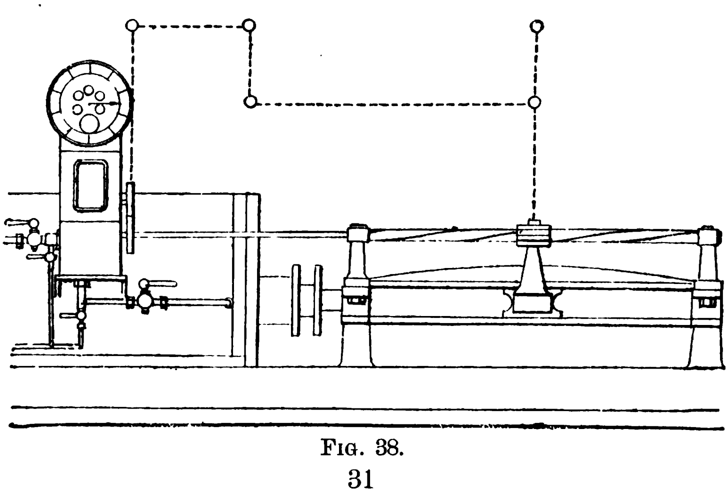


FIG. 38.

Take another reading of the high pressure cylinder a', then the total output $x = \dfrac{a + b}{a} \times a'$

In Fig. 38 the movement of the engine piston is transmitted to the indicator by means of a square rod twisted in the form of a spiral, or else by a system of levers (indicated by dotted lines).

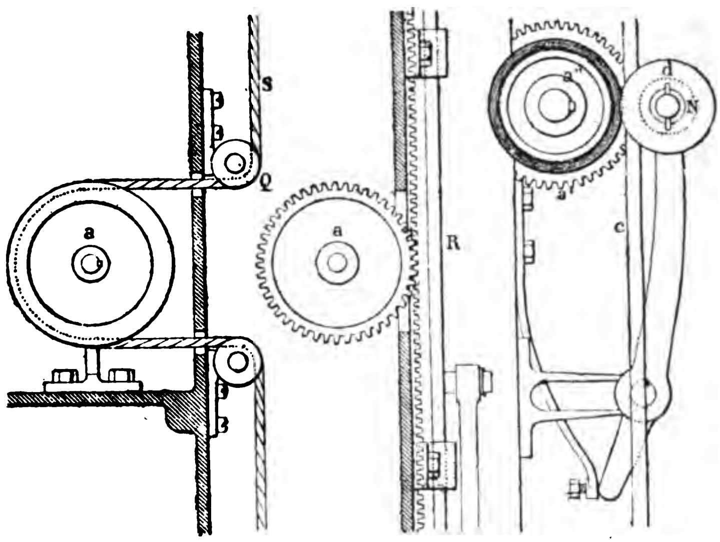


FIG. 39. FIG. 40. FIG. 41.

Figs. 39–41 show the method of driving the wheel a (1) by a cord; (2) by a rack R; (3) by a wheel a', a cylinder or roller a'' and a sliding bar c, acting by friction, and kept pressing upon a'' by d.

A paper drum and a pencil holder may be fitted to the indicator for taking ordinary or differential diagrams.

Calculation of the diameter of the wheel d *in order to give* 1,000 *foot-pounds for each revolution of the pointer* (Fig. 37).

Let p be the pressure which compresses the spring by one inch.

d the diameter of the plunger of the indicator in inches.

q the diameter of the little wheel a.

k the number of teeth on the cylindrical pinion.

l the number of teeth in the wheel.

m the number of teeth in the worm wheel.

n the number of teeth on the first clock wheel.

D the diameter, in feet, of the pulley which it is desired to find.

$\frac{p}{d^2}$ the pressure per circular inch to give a deflection of one inch.

Let us assume that the little wheel a is one inch from the centre of the disc e; one turn of the disc will make the little wheel revolve $\frac{2}{q}$ times. Therefore, for one turn of the disc the pointer will revolve $\frac{2}{q} \times \frac{k}{l} \times \frac{m}{n}$ times.

But, for each revolution of the disc or the pulley the work per circular inch on the engine piston is $\frac{p}{d^2} \times \pi D$ in foot-pounds.

Therefore, for each revolution of the pointer, that is for every 1,000 foot pounds:

$$1{,}000 = \frac{p}{d^2} \pi D \times \frac{q \times l \times n}{2 \times k \times m}$$

from which we obtain

$$D = \frac{2000\,k\,m\,d^2}{3{\cdot}14\,p\,q\,l\,n}\ \text{feet.}$$

H. Lea's Planimeter Indicator (Figs. 42, 43).

This is the best developed combination of the indicator and the planimeter.

FIG. 42.

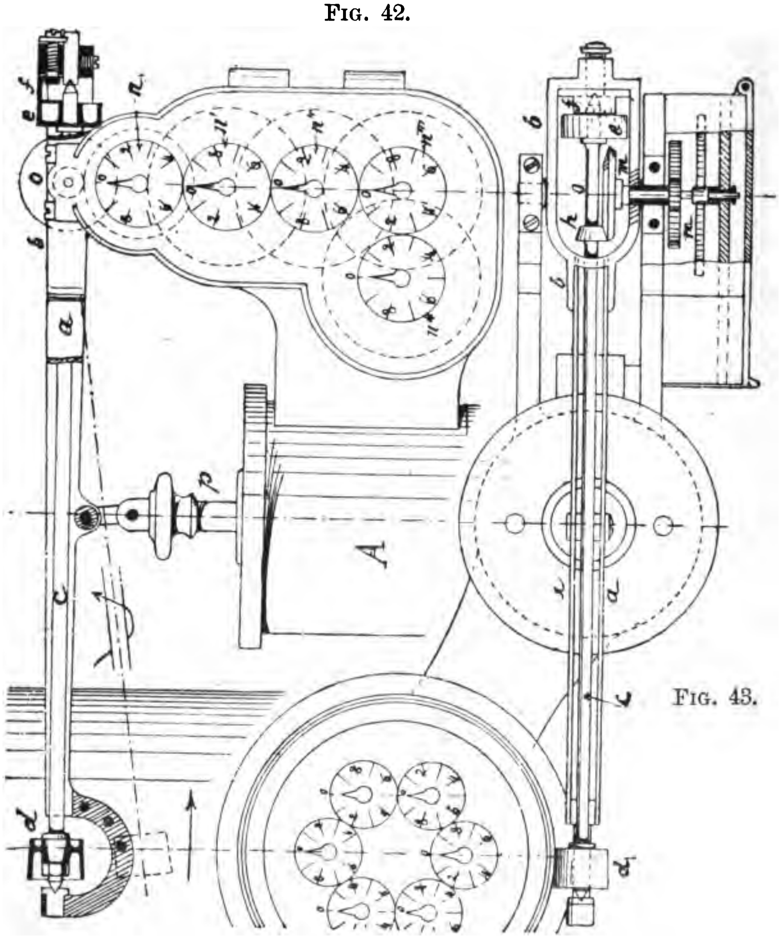


FIG. 43.

The piston p working in the cylinder A actuates the clockwork dials.

The enlarging lever, oscillating at O, is formed of two rods a a, connected to the piston rod p, and ending in a cap b. Between the rods a a is a spindle with a conical steel pinion d at its lower extremity. A toothed wheel e is fitted with a flange and a pin and socket bearing f at its upper end, and keyed upon it is a bevel pinion h. This bevel pinion h actuates a series of dials by means of the bevel wheel m, the first of which (n) registers each revolution of the conical pinion d in units; the next (n') each ten revolutions, and so on.

The number of revolutions made by d is so proportioned that each revolution represents one square centimetre or one square inch.

Theory.

To show that the number of turns of d is proportional to the surface of the diagram, let us assume the diagram to be a rectangle a b c d (Fig. 44). Let d follow the line a b, then b c, and lastly d a.

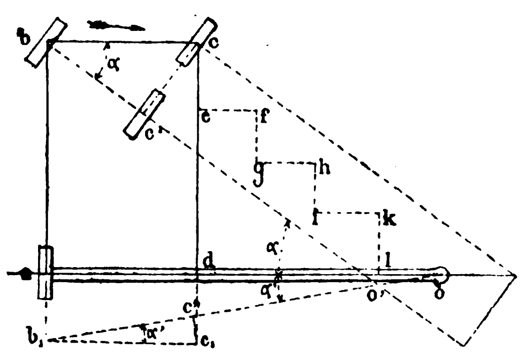


FIG. 44.

The number of revolutions of d whilst rising from a to b will be deducted in descending from c to d, whilst

during the perpendicular travel $d\ a$ it does not revolve; but during the travel $b\ c$ there will be slipping and rotary movement as if the course were $b\ c'$ in slipping and $c'\ c$ in turning perpendicularly on $b\ c'$.

Thus $c'\ c = b\ c\ \sin\alpha$; but $\sin\alpha = \frac{ab}{L}$: L being the length of the oscillating lever $a\ o$.

Therefore $c'\,c = \frac{a\,b \times b\,c}{L}$; but $a\,b \times b\,c$ is the equivalent of the area of the diagram.

Let n be the number of revolutions made by d between c' and c, and let r be the diameter of d.

Then $c'\,c = \frac{a\,b \times b\,c}{L} = 2\ \pi\ r\ n$, from which the area is represented by $a\ b \times b\ c = 2\ \pi\ r\ n\ L$. In the same way, if d follows on the return swing the line $d\ c\ b\ a$, it will turn in the same direction as before from $c\ c''$, and $c'\ c \times c\ c'' = 2\ \pi\ r\ n\ L$.

If we suppose that d follows the whole series of rectangles $a\ b\ c$, $e\ f$, $g\ h$, $i\ k$, its revolution will always be proportional to the sizes of the angles or to the areas.

It will be the same for a series of very small rectangular elements or finally for any curve either above or below the atmospheric line $a\ o$.

To make use of the Indicator.

Set the dial pointers at zero. Measure on the drum the length of the diagram and calculate the proportion which it bears to the travel of the piston.

The scale of the spring is known, and the work represented by each square inch of the diagram can therefore be calculated. The drum is fitted with

Darke's arrangement for starting it at any given moment. Six dials inside the drum register the number of revolutions of the engine.

Suppose the dials are turning in the forward direction and indicate at the moment of stopping 5·627, that figure represents 5·627 square inches. If

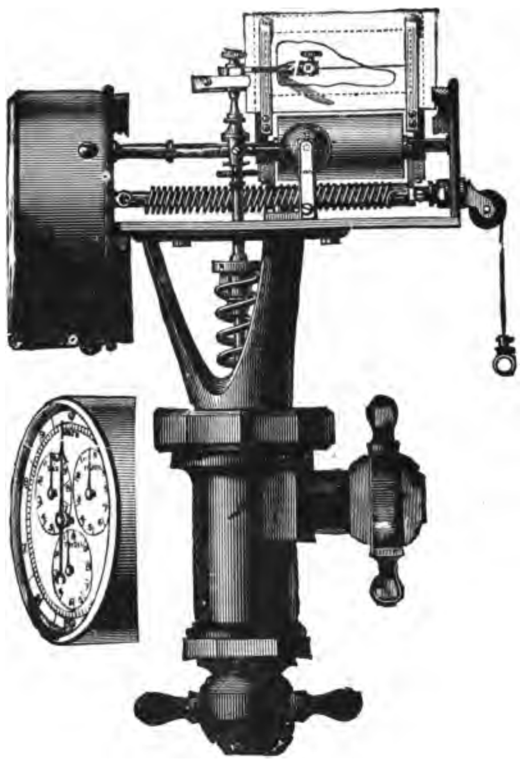
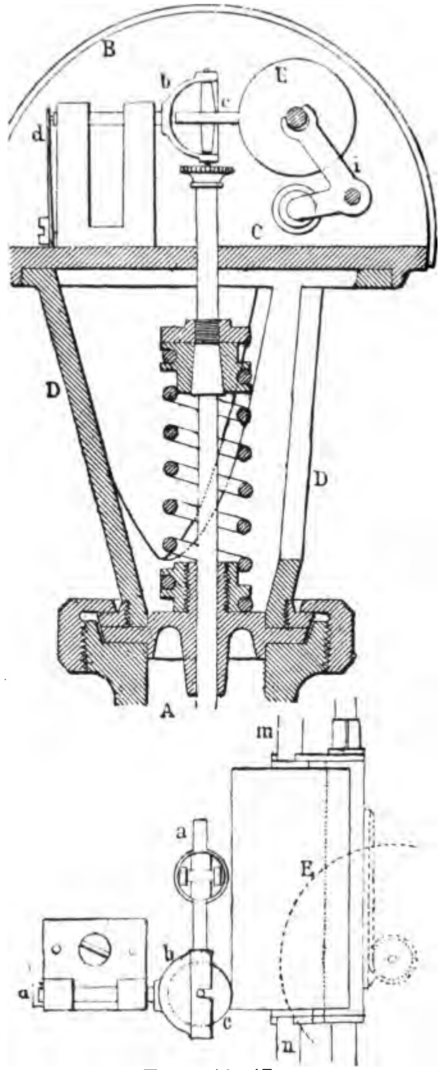


FIG. 45.

on the other hand the dials are turning in the backward direction, and they indicate the same figure on stopping, the area will be 10·000 − 5·627 = 4·373 square inches.

The horse power is that developed on one face of the piston only.

C. Vernon Boys' Indicator (Figs. 45–47).



Figs. 46, 47.

This, like the former, is a combination of the planimeter and the indicator.

Each end of the cylinder of the indicator is connected to each end of the engine cylinder. The spring is electro-plated to prevent it from oxidation.

The oscillation of the plungers *A* are proportional to the effective steam pressure acting on the piston of the engine. This pressure, multiplied by the travel of the piston, enables us to find the horse power at the time of the test. The sum total of the readings gives the total work done.

Integrating Mechanism.

The plunger rod ends in a ball and socket joint, which holds the lever *a b*. This lever is formed of a

half spherical cap *b* to which *a* is attached, and within which is a little wheel *c*. The cap *b* with the lever arm *a*, is free to oscillate on the axis *b d*. A spring *d* keeps *c* constantly pressing on the drum *E*. The drum is free to slide on its axis *M N*: a cord *K* attached to *i* and a return action spring *Z* impart to it a to-and-fro motion proportional to the travel of the piston of the engine; but if the drum tends to turn on its axis, then the spindle *m n* moves with it.

The Action of the Indicator (Figs. 46, 47).

So long as the piston is balanced the work is zero. The arm remains horizontal (*a b*); *c* revolves by contact with the sliding cylinder *E* and conveys no rotary movement to it; but if the plunger of the indicator rises or falls *C* takes an angle in line with *a b*, and owing to the long distance movement of *E* and the pressure of *C* upon it *E* will begin to revolve.

Let us suppose (Fig. 48) that *a b* makes an angle

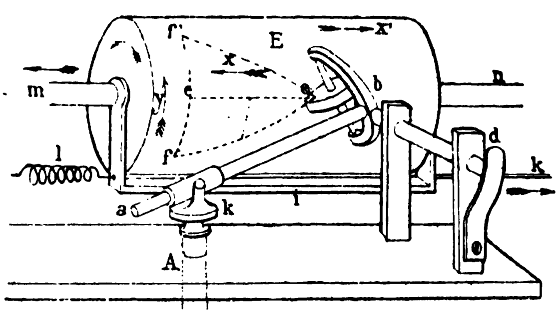


FIG. 48.

a with the horizontal. If the drum *E* follows the line *c e* in the direction of the arrow *x*, the wheel *c* whilst revolving will trace the helix *c f* parallel to *a b* and the drum *E* will have revolved from *c* to *f* in the direction of the arrow *y*.

In the triangle $c\ e\ f$, $e\ f$ is proportional to the effective steam pressure and $c\ e$ to the travel of the piston of the engine. Therefore $c\ e \times e\ f =$ the work for one stroke of the piston. But $c\ e$ is a constant. Therefore the horse power is shown by $e\ f$, the variations of which only need be recorded.

In the return direction x', $a\ b$ may be parallel to $f'\ c$, and the work will be shown by $e\ f'$, whilst the drum revolves in the same direction as before.

Note.—It is not necessary for $a\ b$ to be strictly parallel with $m\ n$ to start with, for if there is any error and there is any exaggeration of $e\ f$ there will be an equivalent diminution of $e\ f'$. To sum up, the total horse power hours will be shown by the numbers of revolutions of the spindle $m\ n$ shown on the dials.

The work in foot-pounds T (corresponding to each unit on the dial) is found by the following formula, which is similar to that we have already described :—

$$T = n\left(\frac{C}{c}\, D^2\, K\, A\right)$$

Where $n =$ the number indicated on the dials,

$C =$ the travel of the piston of the engine,

$c =$ the travel of the drum E,

$D =$ the diameter of the engine piston in inches,

$K =$ the variable coefficient of the springs used,

$A =$ the coefficient for each indicator.

To find the total horse power, therefore, it suffices to calculate the value of the expression shown in the brackets and to multiply it by n.

The indicator can be constructed so as to trace either a true diagram or an ordinary diagram.

CHAPTER III

The Mounting of Indicators

THE indicator should be fitted in such a position —either vertical or horizontal—as to be most readily handled. In the case of high speed engines, in particular, it should be connected directly to the cylinder. The stop and blow-off cocks permit of its being dismantled without stopping the engine.

Fig. 49 shows the stop-cock *R* with three pipes in order that diagrams may be taken one after another at each end of the cylinder. This stop-cock (Fig. 50)

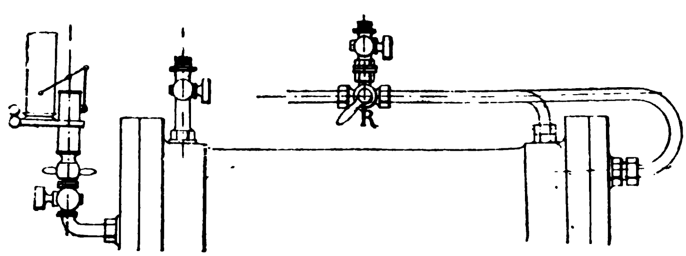


Fig. 49.

is fitted with an outlet *a* to drain the cylinder of the indicator, and the opening of which enables us to trace the atmospheric line.

The pipes, which should have rounded elbows, should also be from a half to five-eighths of an inch in diameter.

The pipes leading from the cylinder must be so placed that the piston does not close them at the end of the stroke.

The indicator must not be mounted on a steam pipe, because, owing to the rapid flow of steam in the pipe, there would be a tendency to depress the plunger and give an unreliable diagram.

Joints should be made with cotton yarn, tallowed, but without red lead, for the smallest particles of this

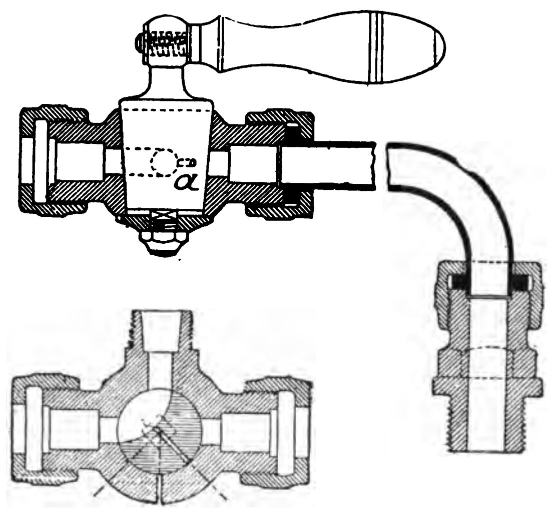


FIG. 50.

in the pipes or the indicator itself would affect its accuracy.

There are various ways of connecting the paper drum with the engine piston. One of the simplest (Fig. 51) consists of two levers, both carried on one spindle, of which the longer is connected at its other end with the sliding crosshead of the engine, and the shorter with a semicircular pulley.

It is essential that the oscillations of the levers should be tangential to the paper drum, in order

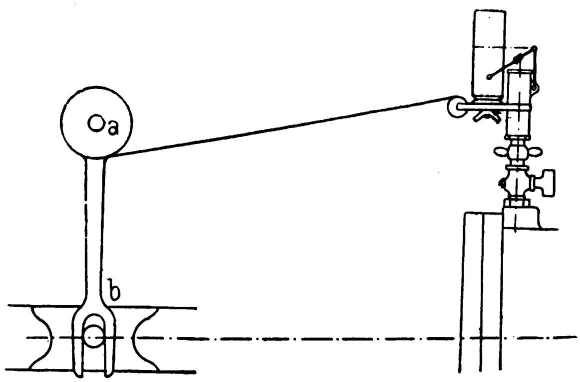


FIG. 51.

that the abscissæ in the diagram may be proportional to the travel of the engine piston.

If the sliding crosshead is not accessible, as in the case of enclosed engines, an eccentric wheel must be fixed to the end of the shaft.

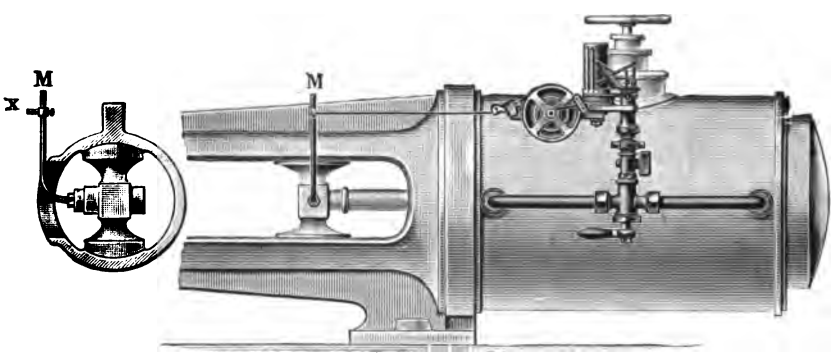


FIG. 52.

Fig. 52 shows the case in which the cord is attached to a bar *M* and the indicator itself is fitted with reduction gear. Fig. 53 shows two indicators,

the first actuating the second, with reducing gear *D R* fixed to the frame of the engine. The cord is

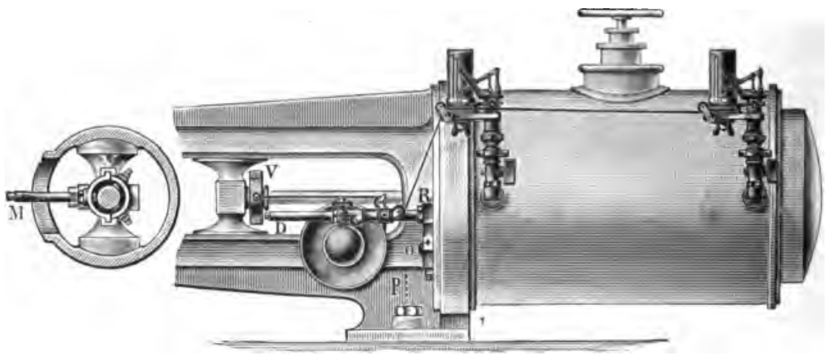


Fig. 53.

attached to the bar *M*, which is fixed to the piston rod by means of the collar *V*.

Figs. 54, 55, 56 and 57 show a method employed when the cylinder is of considerable diameter. A bolt *a* is

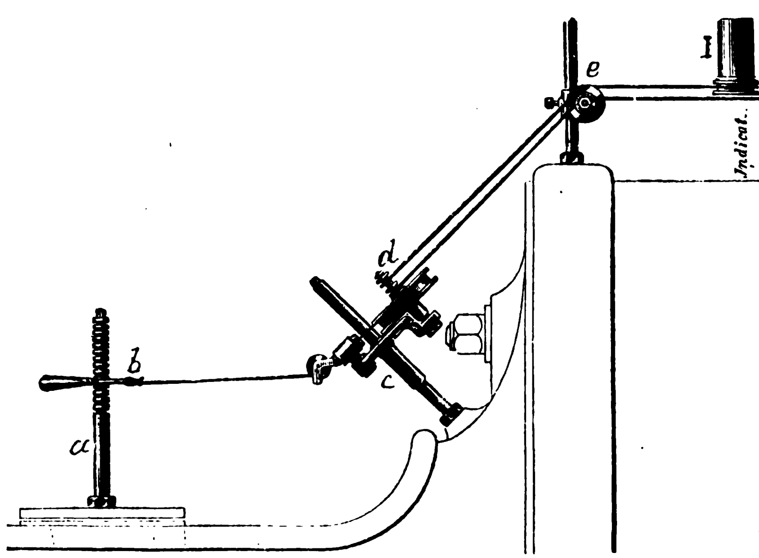
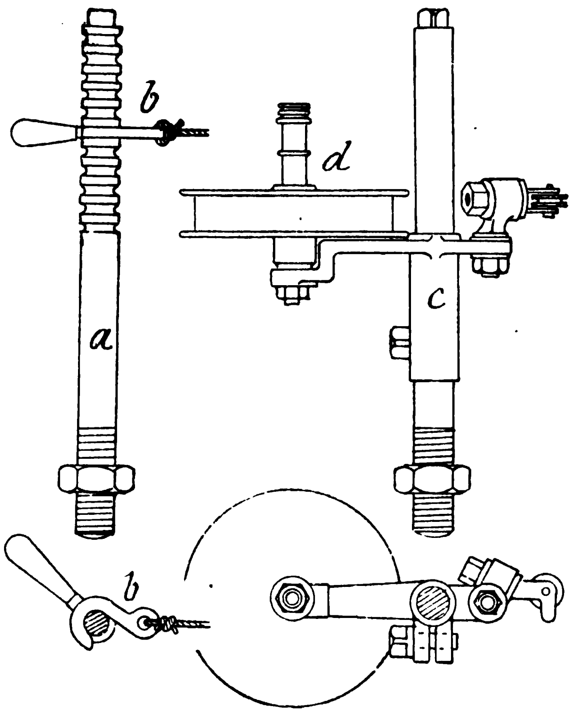


Fig. 54.



FIGS. 55, 56.

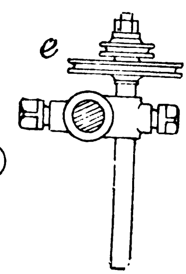
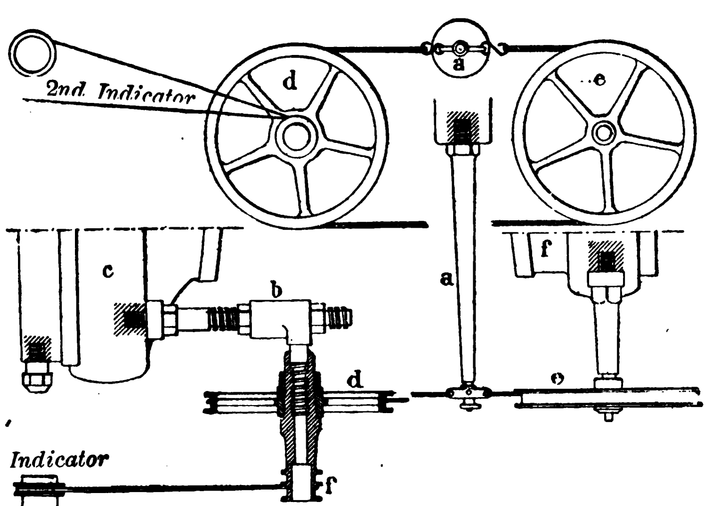


FIG. 57.



FIGS. 58, 59.

fixed to the sliding crosshead and a hook *b* connected to the cord catches into one or other of its grooves in such a position that the cord is parallel to the piston rod. Bolt *C* fixed to the framework of the engine carries two reducing pulleys. The cords starting over these pulleys pass over the pulleys *e*, and each is then conducted to an indicator. For Figs. 52–54 we are indebted to Messrs. Dreyer & Co.

Figs. 58, 59 show another way of mounting two indicators. The bolt *a* screwed to the crosshead carries an endless cord which passes over the pulleys *d* and *e* carried on axles fixed to the framework of the engine. The tension on the cord is adjusted by means of the screw sleeve *b*.

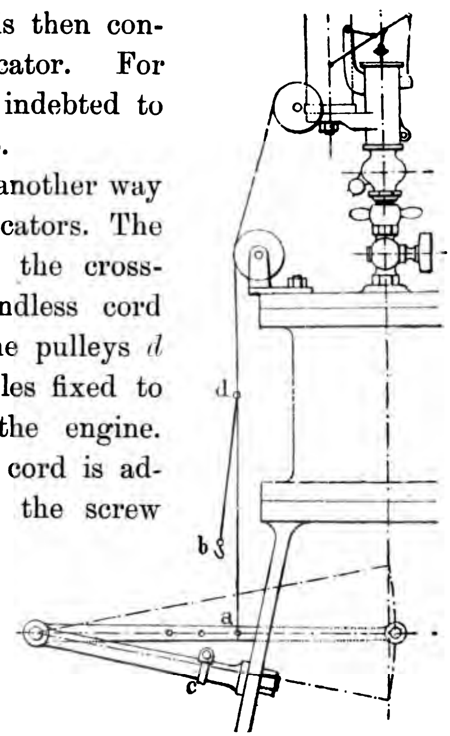


Fig. 60.

The pulley *d*, by means of the internal screw thread, moves sideways at each revolution to and fro, so that the coils of the cord always lie flat on the face of the pulley. The two small pulleys *f* on which the indicator winds and unwinds are attached to *d*.

In the case of a vertical engine (Fig. 60) the lever *a* is fixed at one end of the crosshead and at the other to the fixed bolt *c*. The cord attached to *a* leads to

the indicator, which is here of the Martin Garnier pattern. To stop the indicator it is only necessary to hook *b* into the stationary bolt *c*.

In the case of an oscillating engine (Fig. 61) two methods may be employed, both of which are shown in our illustration. On the right-hand side of the diagram the cord attached to the crosshead passes over the larger reducing pulley and the smaller actuates the indicator *A*.

On the left-hand side the crosshead carries a socket of square internal section which as it moves to and fro causes a twisted square rod to which is affixed the pulley *f* to revolve. The indicator cord is worked by this pulley.

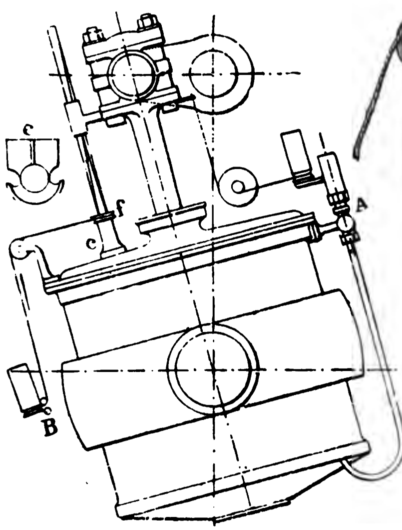


FIG. 61.

FIG. 62.

Travel Reducing Gear.

Stanek's Gear (Figs. 62, 63). The large pulley

fitted with an internal return action spring, and the little pulley which may be changed at will so as to obtain any desired ratio between the crank throw and the movement of the paper drum, are mounted on one and the same screwed bolt and may be adjusted to any desired position on a bolt shown upright in the

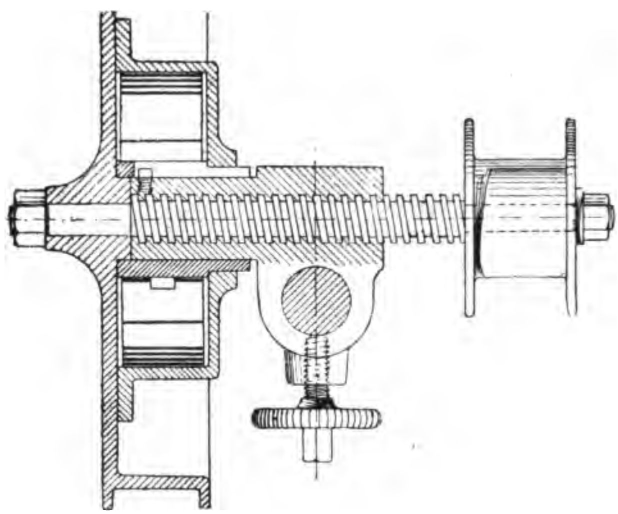
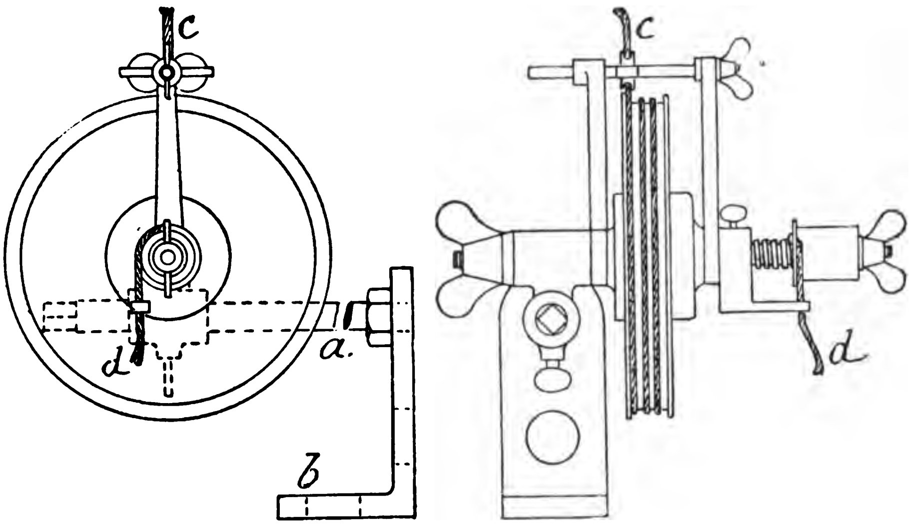


FIG. 63.

illustration by means of a set screw. This bolt, which may be straight or bent as desired, is screwed into a collar, fitted with three fixing screws so that it can be attached to any hexagonal nut on the cylinder or such other suitable support.

An arm, also movable on the bolt, carries the pulleys over which the cords pass to the engine cross-head and to the indicator. These pulleys have a side movement as in Figs. 58, 59, in order that the cord may be flat upon them.

Figs. 64–66 show another method.



FIGS. 64, 65.

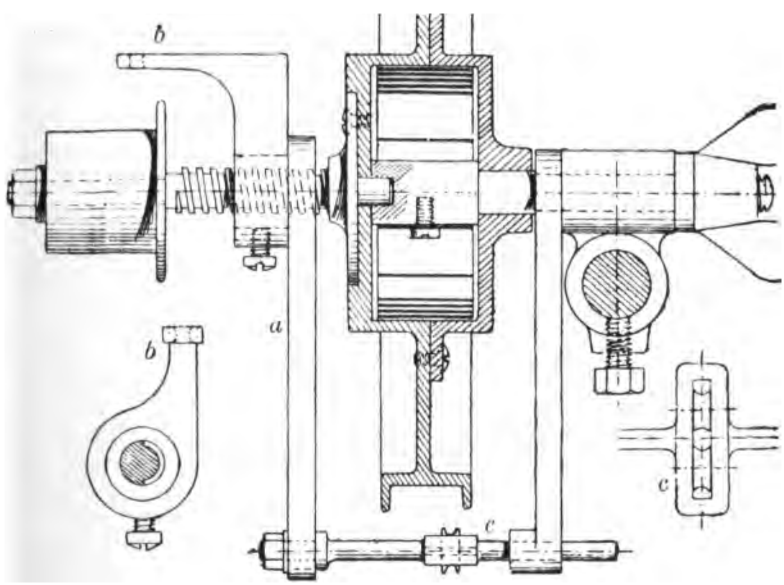


FIG. 66.

The bolt *a* (Fig. 64), which may be either vertical or horizontal as desired, is screwed to the angle iron *b*, which is in turn attached to the frame of the engine.

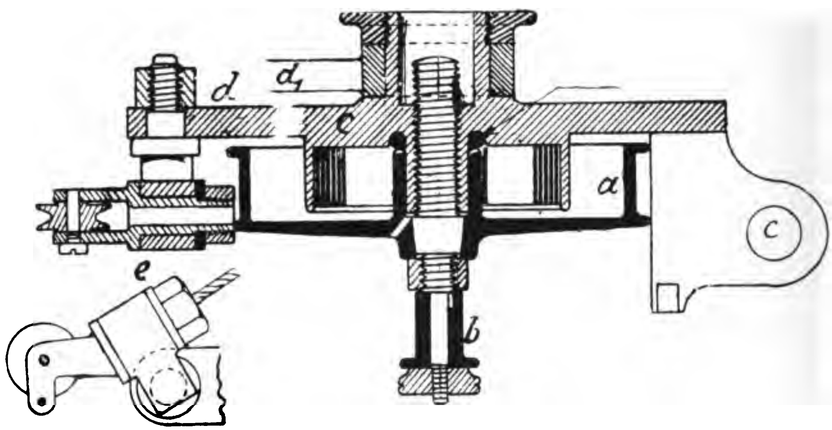


FIG. 67.

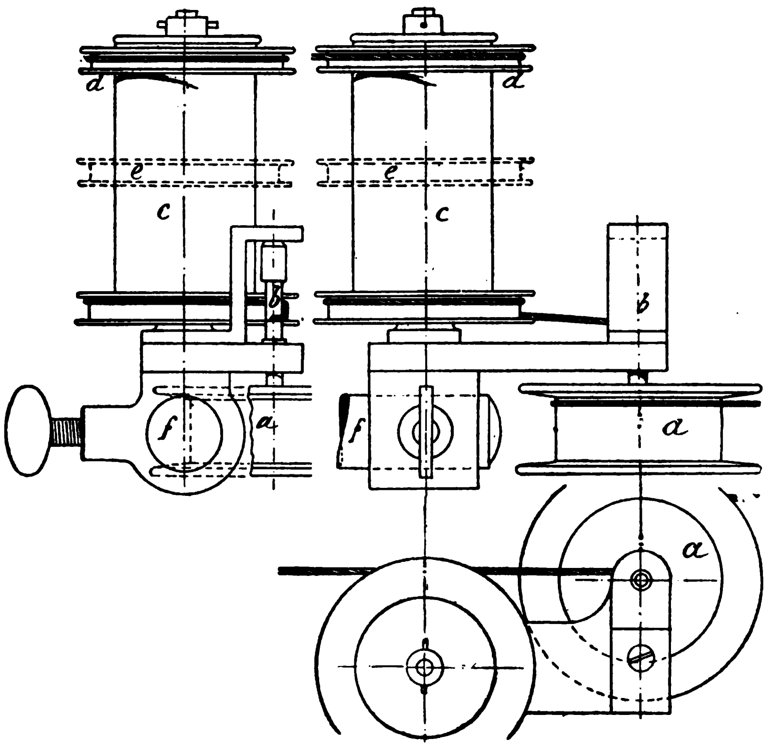
c is the cord that connects the piston to the indicator.

Fig. 66 shows the return action spring inside the large pulley.

Fig. 67 shows the apparatus as fitted to the base of the paper drum itself.

P. Garnier's Apparatus
(Figs. 68–70).

The cord from the engine is wound on the wooden pulley *a*. *b* on the same spindle receives the cord from the cylinder *c*, within which is a return action spring. The cord on pulley *d* works the indicator. A second pulley *e* may be used to work a second indicator. The whole is carried on the bolt *f* attached to the framework of the engine as described above.



FIGS. 68–70.

To Calculate the Diameter of the small Pulley.

Let L = the stroke of the engine.
l = the breadth of the diagram.
D = the diameter of the large pulley.
d = the diameter of the small pulley.

These diameters include the thickness of the string, which is generally about one-tenth of an inch.

We have:

$$\frac{d}{D} = \frac{l}{L}$$

$$\therefore\ d = D\frac{l}{L}$$

Suppose we have D equal to six inches, and l equal to five inches, then

$$d = \frac{30}{L}$$

where L and d are in inches.

The following table gives the values of d for various values of L:—

Piston stroke in inches	24	36	48	60	72
Diam. of winding d	1·25	0·83	0·63	0·50	0·42
Diam. of pulley $(d-0{\cdot}1)$	1·15	0·73	0·53	0·40	0·32

With these five pulleys the length of the diagram for any stroke will be less than five inches, and will be given by

$$l = L\frac{d}{D}$$

Crosby's Reducing Pulleys (Fig. 71).

Here the reducing pulleys are mounted on the same frame as the indicator itself. The large pulley, receiving its impulse from the engine, drives the little pulley by means of bevel gearing. The return action spring is fixed to the spindle of the little pulley; it is helical in form, and contained in a cylindrical covering.

Electric Control of the Pencil.

The object of this is to enable diagrams to be taken on a number of different cylinders at the same moment.

The full lines in Figs. 72, 73 show the electric

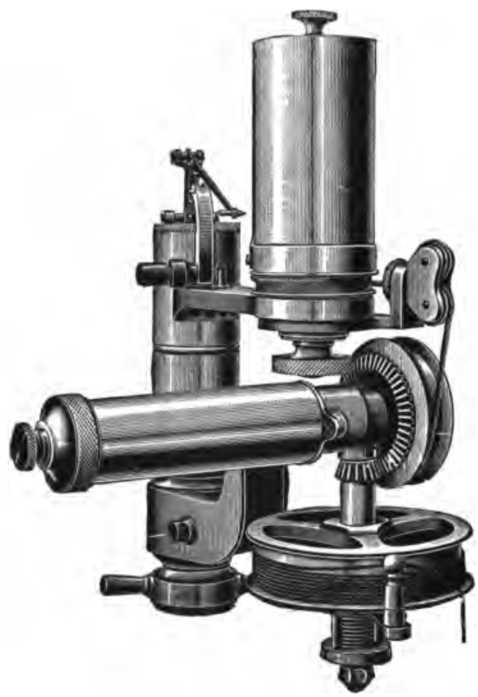


Fig. 71.

connections applied to the Rosenkranz indicator.

The steel plate A is attached to the cap which carries the parallelogram by a fork-shaped piece of steel g fixed to the boss T by two screws.

An electro magnet E is attached to the body of the indicator by a strap R. The distance between A and

the electro magnet is adjusted by the screw *d* and a spring *F*.

Fig. 72.

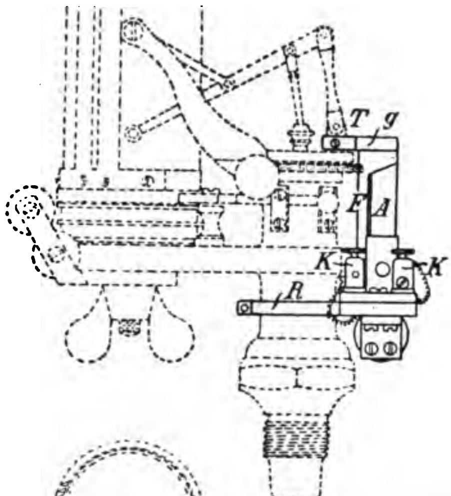
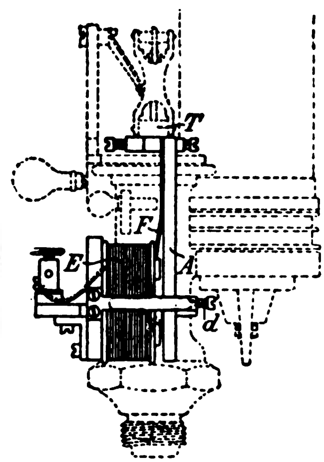


Fig. 73.



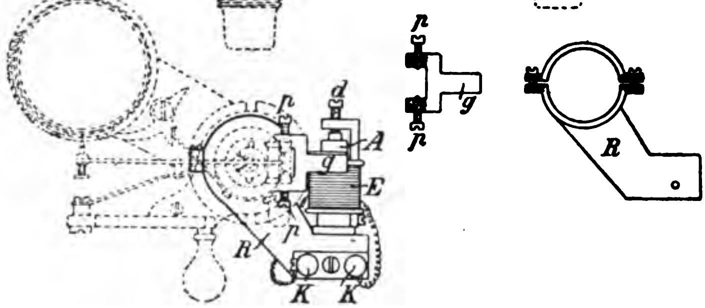


Fig. 74.

K K are terminals of the electric circuit.

On closing circuit the electro magnet attracts *A* and gives such a movement to the parallelogram as to bring the point of the pencil into contact with the paper on the drum; on breaking circuit the spring pushes back *A* and lifts the pencil.

Fig. 75 shows the connections for a circuit to two or four indicators, the switch being indicated by 1.

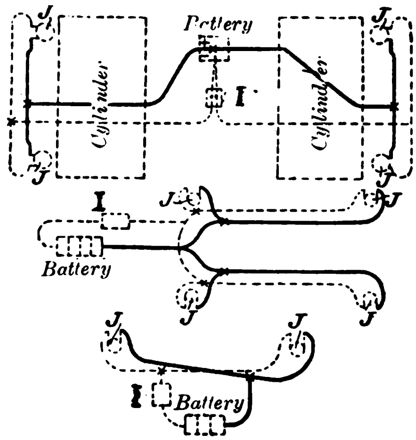
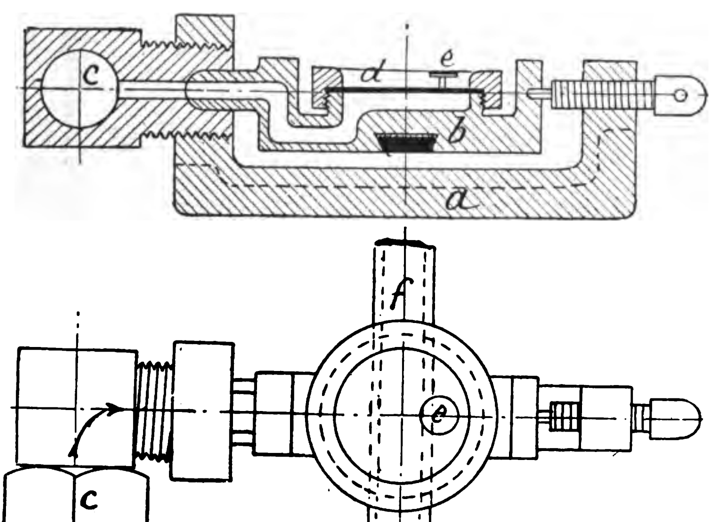


FIG. 75.

Professor John Perry's Optical Indicator (Figs. 76-78).

Indicators with springs become erratic in their



FIGS. 76, 77.

action when the engines under test are of the highest speed class. Only when the period of oscillation of

the indicator is less than $\frac{1}{20}$ that of a revolution of the engine is the diagram satisfactory. At $\frac{1}{15}$ the diagram is unsatisfactory, at $\frac{1}{10}$ it is defective. The optical indicator consists of a fixed portion *a* (Figs 76, 77) carrying a box *b* by means of two

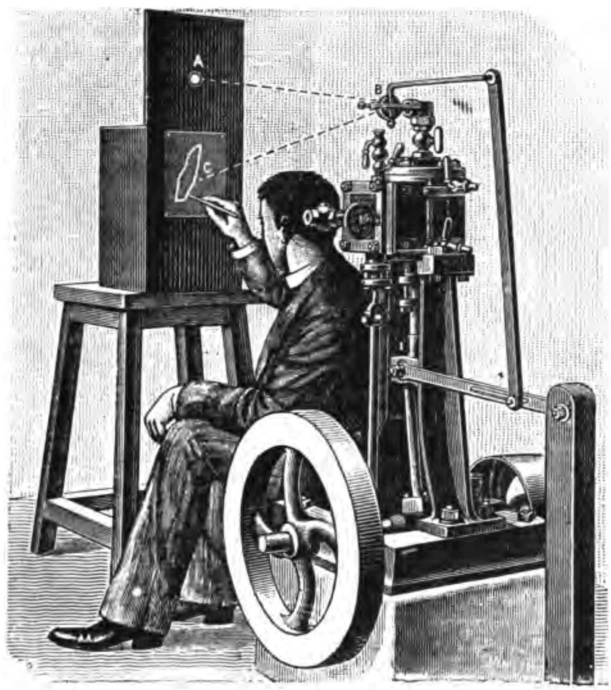


FIG. 78.

pivots; *c* is the steam inlet. The box *b* is closed by a sheet of steel *d*, which, being thin and elastic, yields somewhat under the pressure of steam.

Upon this steel diaphragm, about halfway between the centre and the edge, is fixed a mirror *e*.

A lever *f* connected with the crosshead of the engine causes *b* to oscillate from side to side at each stroke.

The rapidity of the oscillations of the diaphragm may easily be 500 per second or even more, so that correct diagrams may be obtained at the highest speeds.

Fig. 78 shows the indicator fixed to an engine.

A ray of light is thrown by a lamp through *A* on to the mirror, and is projected back on to a piece of ground glass, on which the diagram is shown.

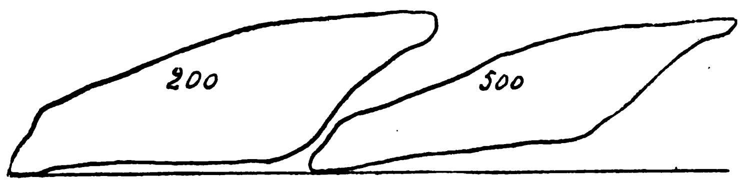


FIG. 79.

Even at a speed of only 100 revolutions the impression is clear enough to trace the diagram by hand with a pencil either on the ground glass itself or on tracing paper fixed over the glass.

To trace the atmospheric line, open *c* to the air; when this has been done, admit steam, and a second line will be traced; the distance between the two will give the pressure scale of the diagram.

Fig. 79 shows two diagrams taken, one at 200 revolutions of the engine, the other at 500 revolutions.

CHAPTER IV

THE THEORY OF THE INDICATOR

BY theory is understood the part played by each portion of the indicator.

The Plunger.

See that in pushing the plunger to and fro by hand there is no appreciable friction between it and the inside of the cylinder.

A loose plunger allows an escape of steam, of little importance in the case of non-condensing engines; but in the case of a condensing engine air passes and destroys the vacuum under the plunger, especially if the pipes to the indicator are long or of small diameter.

If the indicator is not in good order, the spring under compression may press sideways, and so cause friction on the plunger and plunger rod. To prevent this Mr. Lyne attaches the spring to the plunger-rod by means of a ball-and-socket joint (Figs. 80, 81).

Tests—the Scale of the Spring.

Turn the apparatus upside down and fit it in a vice; then compress the spring by means of weights affixed to the plunger-rod.

Turn the paper drum by hand, and the various deflections will be recorded by the pencil.

The indicator may also be fixed to a bracket or

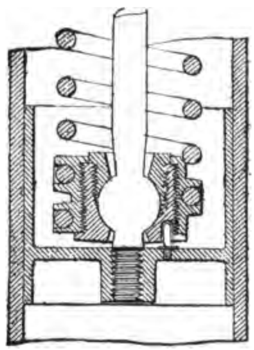


FIG. 80.

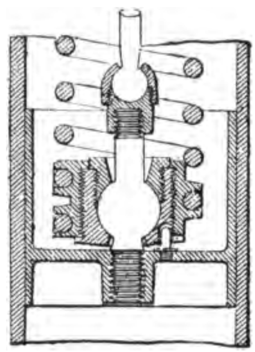


FIG. 81.

such a device as shown in Fig. 83, where the rod carrying the weights is connected to the plunger-rod by means of the stirrup Z, and passes at its lower extremity through the guide hole B.

If the spring has to be tested for expansion, in order to measure pressures below the atmospheric line, the indicator must be fixed upright and above the cross piece *R*.

Let q denote the pressure in pounds for every tenth of an inch of deflection.

s the surface of the plunger per square inch.

Then $\frac{q}{s} = p = \begin{cases} \text{load per square inch for every} \\ \text{tenth of an inch deflection.} \end{cases}$

The scale of the spring $e = \frac{1}{p} = \frac{s}{q}$.

The flexibility of a helical spring, within the limits for which it is designed, is proportional to the pressure upon it. If (Fig. 82) the abscissæ repre-

sent the pressures, and the ordinates their corresponding deflections, it will be found that the line

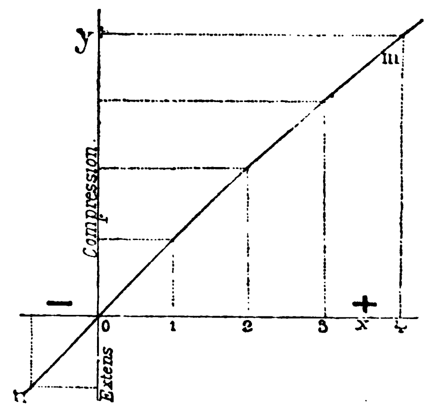


FIG. 82.

indicating the relation of the one to the other is for all practical purposes a straight line.

The same is the case if we consider the effect of vacua as negative pressures and continue the line from *o* in the direction *o n*.

If pressures have been applied successively to 15, 30, 45 pounds per square inch on the piston surface, and the corresponding deflections measured $f\,f'\,f''$, we still have $e = \frac{f}{15} = \frac{f'}{30} = \frac{f''}{45}$ = the scale of the spring.

In the arrangement (Fig. 84) the spring alone is placed in a cylinder *A* (Fig. 85) built up of two sections and bolted together. The parallelogram *D* traces the deflections on the scale *T*.

In these tests the weight of the rod *K* must be taken into account.

To test the spring under expansion, the glands at

the two ends of the cylinder must be exchanged, and the spring must hang from the upper one.

The Hot Test.

The indicator spring being in connection with the open air, it is always at a lower temperature than that of the steam in the engine cylinder, and as moreover the temperature of the steam varies from the time of admission to that at which it exhausts, we may assume that the working temperature of the spring is but little more than 212 degrees Fahrenheit.

In order to test under working conditions, a little boiler heated by gas or oil is connected up to the apparatus (Figs. 83, 84). In Fig. 83 the steam is admitted by the cock *P*. In Figs. 84, 85 the steam enters the cylinder at *A* through the pipe *p′*, and the condensed water escapes at *p*.

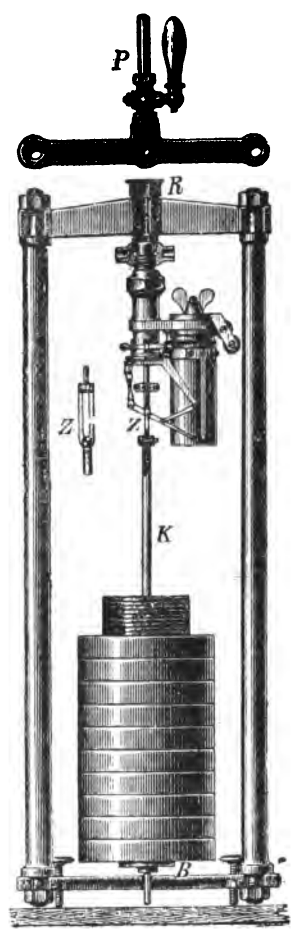


FIG. 83.

In comparing the deflections shown on the diagram (Fig. 82) with those obtained with the apparatus shown at Fig. 84, allowance must be made for friction, and also, what is more important still, the ratio between the altered positions

of the plunger and the pencil. (This point will be dealt with later on.)

In the preceding tests, when the full pressure has been reached, the steam cock must be closed; and the steam gradually condensing, the pressure falls through

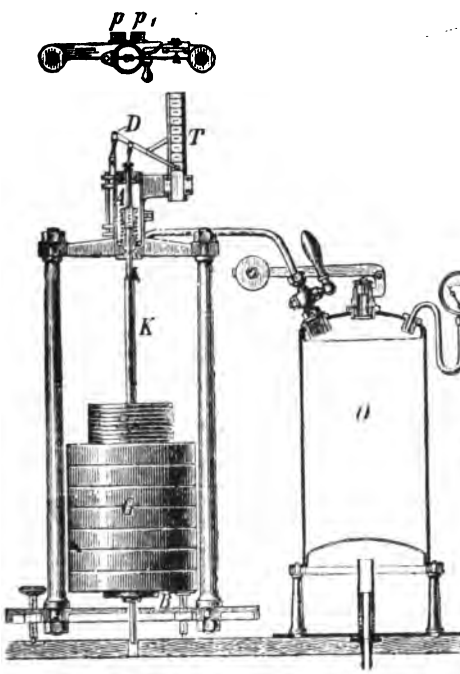


FIG. 84.

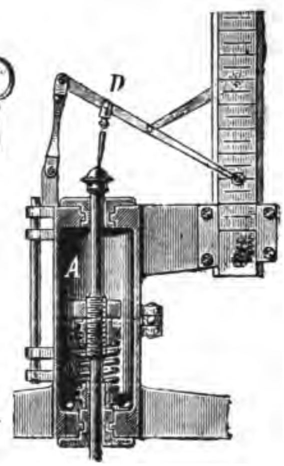


FIG. 85.

the same scale as in rising, and the pencil should give the same readings during the fall of pressure as it recorded when steam was being admitted.

These tests may be applied equally well to the testing of vacua.

Relation between the Movement of the Plunger and that of the Pencil.

The parallelogram must work freely, but there must not be any unnecessary play. In order to see that it works freely, compress the spring and let it expand again and trace a line on the drum which may be turned by hand. Then extend the spring, and as it returns to the normal position let it trace another line. These two lines ought to be identical. Be sure that the movement of the pencil throughout is proportional to the movement of the plunger.

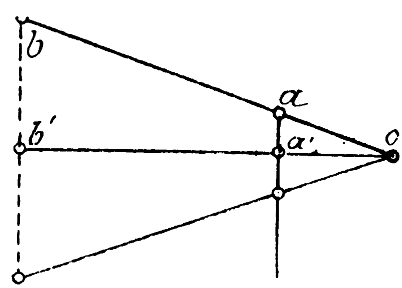


FIG. 86.

In the case of Darke's Indicator, of which Fig. 86 is the diagram, the line b - b' traced by the pencil is straight owing to the guide through which the pencil runs.

The ratio of the travel of the plunger to that of the pencil is maintained constant, for in all the triangles formed by the points $o\ a\ b$ we have, $a\ a' : b\ b' :: o\ a : o\ b :: o\ a' : o\ b'$.

In the case of the Evans' parallelogram (Fig. 87) the

best form, and the one in which the pencil traces the straight line, is his first design in which the connecting rod $e\ d = \frac{1}{2}\ b\ c$. When $e\ d$ is smaller, the line traced by

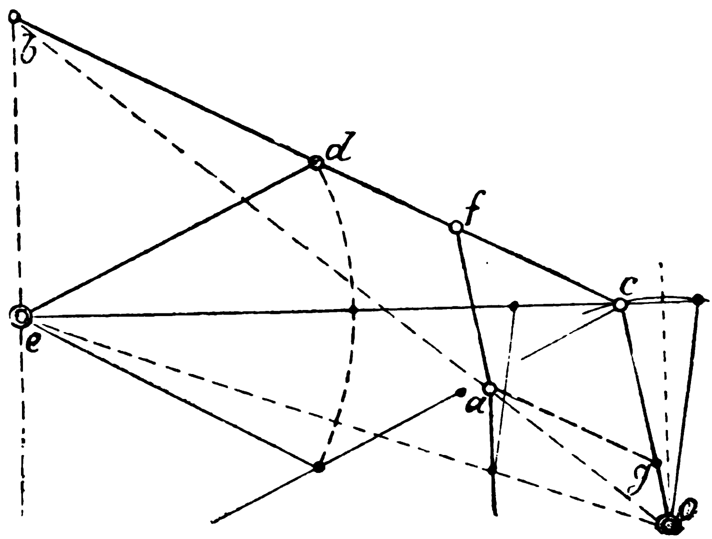


FIG. 87.

the pencil is not so straight. The ratio of the displacement of the plunger a to the pencil b is found by

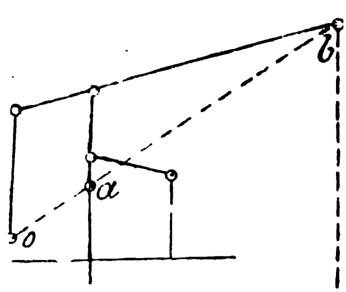


FIG. 88.

taking a point a on $o\ b$; o being the fixed point on

which the rod $o\ c$ moves; the piston rod $a\ f$ being parallel to $o\ c$.

$a\ g$ equal and parallel to $f\ c$ may take the place of $e\ d$; $a\ f$, $c\ g$, $f\ c$, $a\ g$ then form a lozenge pattern and the whole constitutes a pantograph, the points $o\ a\ b$ being always in the same straight line. This applies equally to Crosby's Indicator (Fig. 88).

The Pencil or Tracer.

The pencil used is sometimes of hard lead, but as a

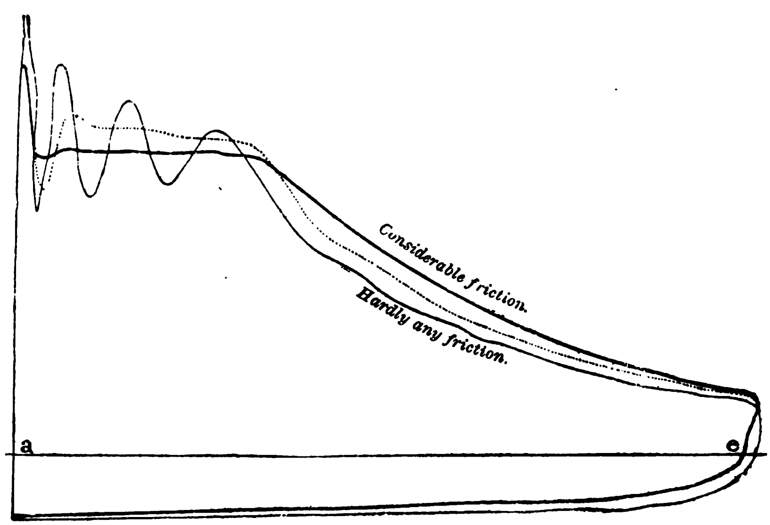


FIG. 89.

rule a rounded metal point is used which traces a grey line on paper specially prepared. The pencil should press on the paper without scratching it. It should be light, for its speed being greater than that of the arms its inertia must tend to set up vibration.

The friction of the pencil on the paper reduces these vibrations, but at the cost of some want of

accuracy in the diagram. Fig. 89 taken with a Kenyon Indicator from a Corliss engine shows this very clearly.

It shows that during the admission of steam the line

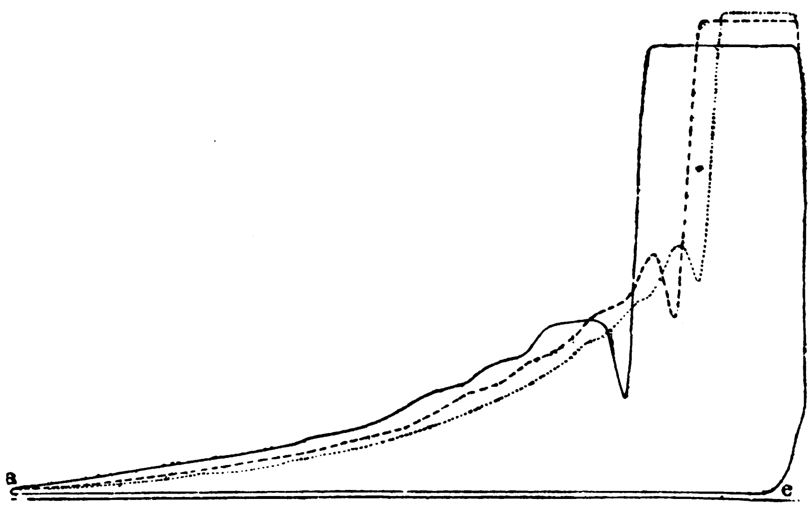


FIG. 90.

of admission is either too high or too low, depending upon whether the vibration of the pencil stops on a down stroke (heavy line) or an up stroke (dotted line).

The friction raises the line of expansion as well as that of exhaust.

When there is friction in the various parts and especially the plunger, in addition to that of the pencil, the curve shows sudden jumps (Fig. 90) due to the jerky movement of the pencil owing to the difference in the coefficients of friction when starting and when in motion.

Vibration of the Pencil.

This is due to the rapid movement of the mechanism set up by the sudden action of the steam and the resistance of the spring. Acted upon in turn by these the plunger moves now to one side, now to the other, of its position of equilibrium and the pencil traces wavy lines.

During expansion the state of equilibrium is best maintained, but during exhaust the oscillations are less marked because the pressure on the plunger is so much smaller.

That the duration of the oscillations is independent of their magnitude is shown by Fig. 91.[1]

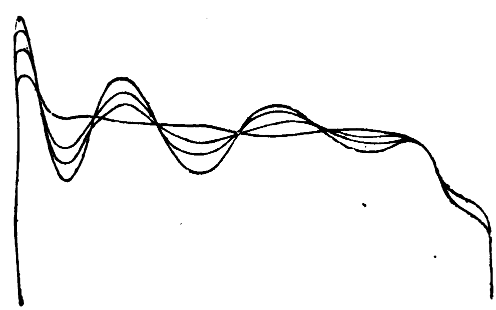


FIG. 91.

The more rapid the oscillations the quicker they exhaust themselves, and there is less fear of them distorting the diagram. This is the reason why indicators in which the compression and expansion of the spring is reduced to a minimum have taken the place of the older forms.

[1] We have to thank M. de Maupeon for the use of the diagrams, 88 to 91.

Formerly endeavours were made to attain the same result by making the sectional area of the steam inlet pipe $\frac{1}{10}$th that of the cylinder, but in this case the effect of the vacuum is interfered with.

Seeing that compression diminishes the jerky action of the steam behind the plunger, it is obvious that to decrease these oscillations springs of greater stiffness should be used in proportion as the compression is less.

The Paper Drum.

The drum should be perfectly round, and the paper laid smoothly on it. Its movement is transmitted to it by a cord made of hemp, catgut or wire, all more or less elastic. At the commencement of a stroke, the

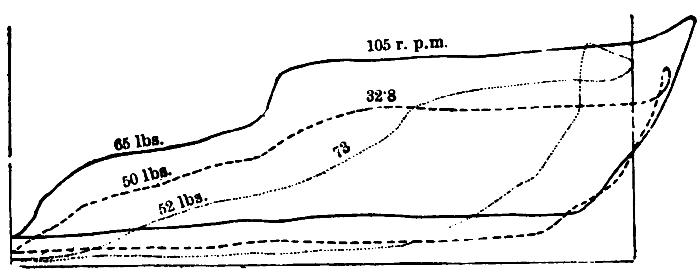


FIG. 92.

cord, in overcoming the inertia of the drum, the reducing pulleys and the tension of the springs, stretches somewhat and the drum is a little behindhand in starting until the cord returning to its normal length ends by giving the drum a movement proportional to the stroke of the engine. There must therefore be a shrinkage in the diagram and a reduction of its area, greater or less in degree as the

speed of the engine is fast or slow or the elasticity of the cord considerable or the reverse. But in the indicating of high speed engines there is often a lengthening of the diagram caused by the impetus of the drum, as shown in Fig. 92.

In any given case this can be overcome by reducing the travel of the drum and increasing the tension of the return action spring in proportion to the speed of the engine.

The Cord.

The cord should be flexible, but should have as little elasticity as possible. It is not infrequently made of

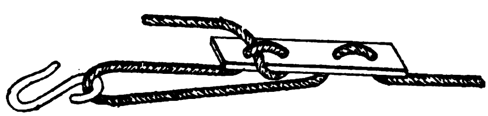
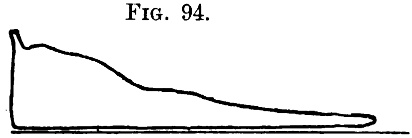


FIG. 93.

hemp or catgut, but it is sometimes made of metal threads or steel strips. A method of connecting the hook to the cord is illustrated in Fig. 93.

A hempen cord should be well stretched before use; it should be dry—for damp increases elasticity—and as short as possible; the pull should be straight and regular without shaking or vibration.

FIG. 94.



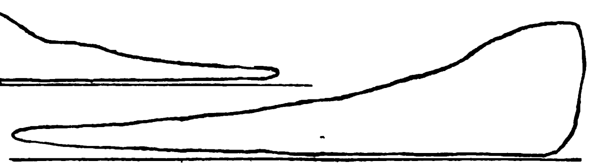


FIG. 95.

Drum worked by Non-flexible Rod.

The two figures shown above illustrate the difference between diagrams obtained when the drum is actuated with a cord and with a non-flexible medium. Fig. 94, which was obtained with a cord susceptible to vibrations and somewhat elastic, gives a less regular curve than Fig. 95 taken with the rigid methods shown on page 14, Figs. 18 to 21.

CHAPTER V

The Study of Diagrams

The Diagram.

BEFORE taking a diagram, admit steam to the indicator for a short time in order that it may become heated to the same temperature as the steam, then shut the steam cock, which action opens the indicator to the atmosphere, and trace the atmospheric line.

Then open the cock again, admitting steam, and if the object is to ascertain the effect of the cut-off and expansion in the cylinder trace one diagram; but if it is to ascertain the mean horse power take several.

At each test note—

1. The constant factors, such as the dimensions of the cylinder of the valves and steam and exhaust pipes and clearance spaces; the scale of the spring and the length of the diagram, and if it has been traced at slow speed.

2. The variable factors, as they exist at the moment when the diagram is taken, namely, the steam pressure at the boilers, the vacuum, the number of revolutions of the engine; indeed all the conditions under which the engine is working and the test made.

In all the following diagrams, *a e* (Fig. 96) is the atmospheric line traced by the pencil before admission of steam; *o x* the line of absolute vacuum, traced below the atmospheric line at a distance equal to 15 lbs. on

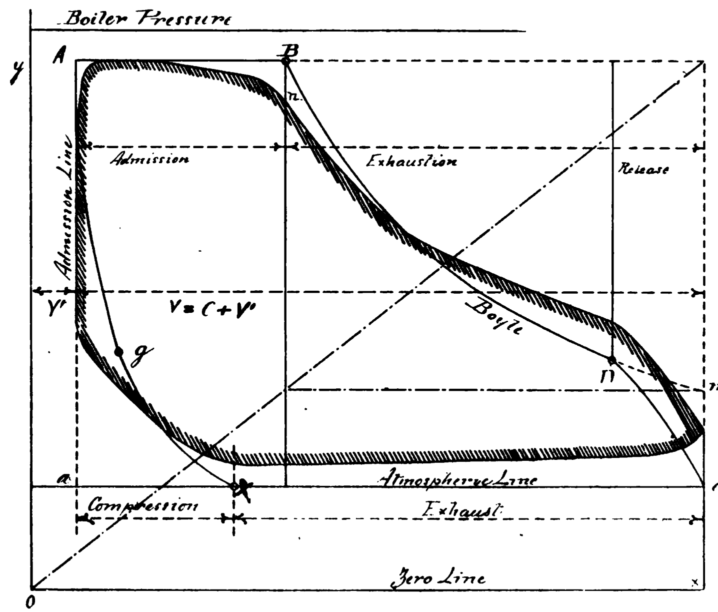


FIG. 96.

the scale of the spring. Strictly speaking this should be 14·7 lbs. for a barometric height of 29·9 inches.

The line *o y* at right angles to *o x* lies at a distance *v′* from the origin and represents the waste spaces in the cylinder which are filled with steam, and which do no useful work.

Each diagram is divided into six periods, namely:

Forward travel.
1. Advance admission.
2. Admission.
3. Expansion.
4. Advance exhaust.

Backward travel. { 5. Exhaust or vacuum.
6. Compression.

In actual practice the diagram shown theoretically by *A B D e f g* (Fig. 96) is much modified, following more closely the shaded outline, the reason for which we will proceed to explain.

Advance Admission.

This is the extent to which the valve is open and admits steam when the piston is at the end of the stroke, its effect being: 1. To take up the play in the connecting rod bearings at the commencement of each

FIG. 97.

stroke. 2. To allow of the full pressure of steam acting on the piston at the commencement of the stroke.

Compression has the same effect; therefore when there is considerable compression there need be less advance of admission and vice versa. If the advance admission is well designed the line of the diagram and *a* coincide at the commencement of the stroke. The small oscillations which are almost always apparent at the top of this line are caused by the impulse of the piston, and vary in greater or less degree as the spring

is flexible or the reverse. The advance admission is too great in the case of *A* (Fig. 97) and there is none in that of *B*; it is insufficient in the case of *C* (Fig.

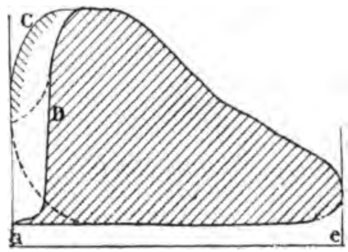


FIG. 98.

98); and in that of *D* admission is late. The dotted line shows the late admission after compression.

Compression partly counteracts a late advance admission and the indicator diagram is the only way in which one can see whether the advance admission is suitably adjusted.

Rapid Action of the Steam.

The time taken by the steam to attain its maximum pressure within the cylinder can be found by cutting

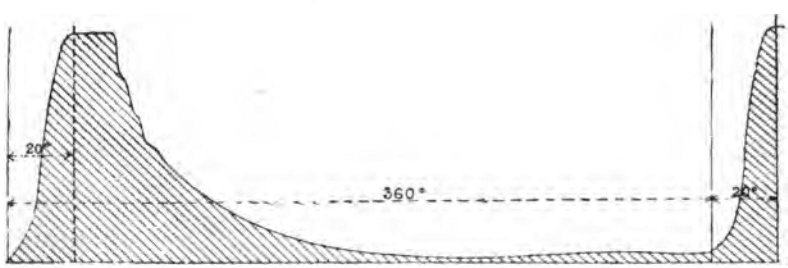


FIG. 99.

off all advance admission and giving the paper drum a movement proportional to that of the crank.

In this way Mr. Vidmann obtained Fig. 99 on a Cor-

liss Engine running at a speed of 60 revolutions with no advance admission, from which we see that $\frac{20}{360} = \frac{1}{18} = 0{\cdot}055$ second, the time taken by the steam in this case to reach its maximum pressure.

Duration of Admission.

If the full pressure of steam is on the piston from the commencement of the stroke, and if the area of the pipes leading to the indicator is sufficiently large, the admission line will be practically horizontal and identical with $A\ B$ (Fig. 96).

The pressure in the cylinder is less than that at the boiler. The fall in pressure increases with each degree of moisture in the steam. It is less marked when the steam pipes are short and straight, and the lower the speed of the steam in the pipes. The size of the pipes is usually calculated for a movement of steam along them at the rate of 100 feet per second. The amount of condensation on admission depends upon the extent of the walls of the cylinder, their tendency to conduct heat and the difference between their temperature and that of the steam. Where the valves and piston rub on the walls of the cylinder their natural conductivity is lessened owing to the lubricant upon them. And the same is the case with other parts where the grease or oil finds a bed.

Condensation goes on until the walls of the cylinder have attained the same heat as the steam; and is least in the case of non-condensing engines and those working without expansion.

To show the amount of condensation arising from extreme differences of temperature, let us take the

case of steam at 6 atmospheres, where $t = 320°$ F. If the free exhaust takes place at 1·2 atmosphere, for instance, then $t' = 222°$, $t - t' = 98°$.

Suppose that it is a question of heating 50 lbs. of cast iron having a specific heat of 0·14.

The heat needed will be

$$98 \times 50 \times 0{\cdot}14 = 686\ B.\ T.\ U.$$

But 1 lb. of steam at 6 atmospheres contains latent heat equal to 888 *B. T. U.*

The weight of condensed steam will therefore be

$$\tfrac{686}{888} = 0{\cdot}76 \text{ lb. nearly.}$$

For any given engine the proportion of steam condensed to that used effectively decreases with increased admission and number of revolutions. Speed is here an important factor.

Enormous waste takes place in the case of low speed engines without lagging, and the waste is often greater in such cases where there is expansion than where there is none.

In his book on screw-propelled steamships, Admiral Paris relates that the *Roland*, steaming without expansion, developed 550 HP. with a coal consumption of 7·98 lb. per HP. hour, whereas with a 0·4 cut-off the same engines only developed 201 HP. with a consumption of 8·8 lb. of coal per HP. hour. With condensing engines showing the same defects it has even been found more difficult to obtain a vacuum when working with expansion than without.

Clark, in his experiments with locomotive engines, has shown that with cut-offs equal to 12%, 30% and 74% the condensation has amounted to 42%, 24% and 11% of the weight of steam admitted to the cylinder.

Condensation is reduced by lagging and specially by steam jacketing.

The speed of the piston increasing from the commencement to half stroke, if the area of the inlet is not large enough, the line of admission falls (Fig. 100).

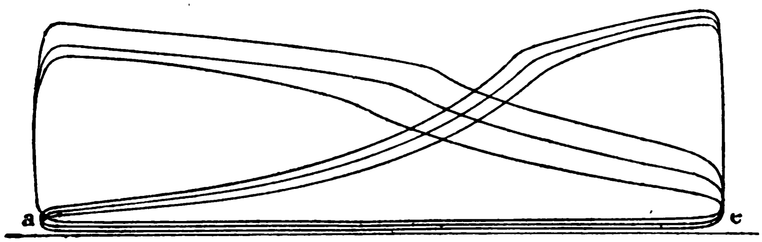


Fig. 100.

This is called wiredrawing. Slide valves gradually closing the ports cause wiredrawing towards the end of the period of admission; and the line of the diagram takes the form shown at 1 and 11 (Fig. 101),

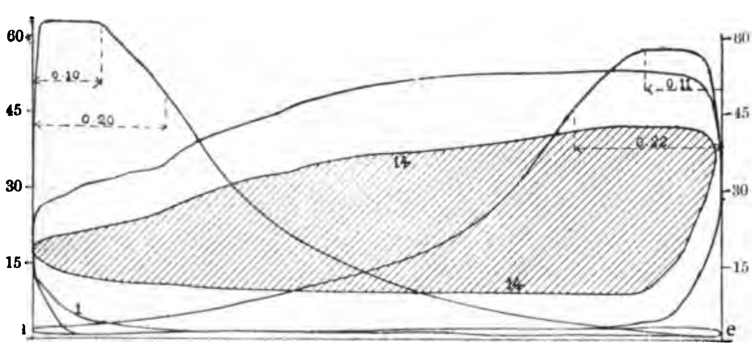


Fig. 101.

where it is seen that admission only completely comes to an end at 2 and 22.

At the highest speed (14 revolutions), and with ports wide open the pressure falls on admission and rises

when exhausting (owing to insufficient area), and the effective work is only equivalent to that part of the diagram which is shaded.

The loss from wiredrawing with ports wide open and gradually closing is shown for a similar cut-off by the triangle *A B n* (Fig. 96), which represents the difference between the theoretical and actual diagram.

The work done per pound of steam is independent of wiredrawing. Steam pressure practically varies inversely in proportion to its density, and the power given out at full pressure being equal to the volume multiplied by the pressure, that of 1 lb. of steam is, for an equal degree of expansion, independent of the initial pressure. This would be truly the case if there were no compression, but the power given out in effecting compression is part of the total power, and is greater in proportion as the initial pressure at admission is less. It is therefore important to have the pressure high on admission and to vary the power by altering the cut-off. In accordance with the properties of saturated steam, the pressure decreased by wiredrawing sets free a certain quantity of heat which produces superheated steam. Thus partial evaporation of such water as has found its way from the boiler may be effected by the steam being wiredrawn.

Expansion.

In the theoretical diagram (Fig. 96) expansion commences at *B*, but in practice expansion begins with the first narrowing of steam inlet, and this is termed for-

ward expansion, the full expansion only commencing at *n* when the steam pressure has become reduced by wiredrawing. There is, therefore, during the closing of the valve a loss of horse power over and beyond that shown by *A B n*. It is often difficult to ascertain where *n*, the commencement of expansion, is really situated; for it is the turning point between the curve of wiredrawing and that of expansion.

After *n*. As the piston moves forward, the pressure falls, following a different law for each engine, and dependent upon its structure and the nature of work it is doing.

Mariotte's (or Boyle's) theoretical pressure curve

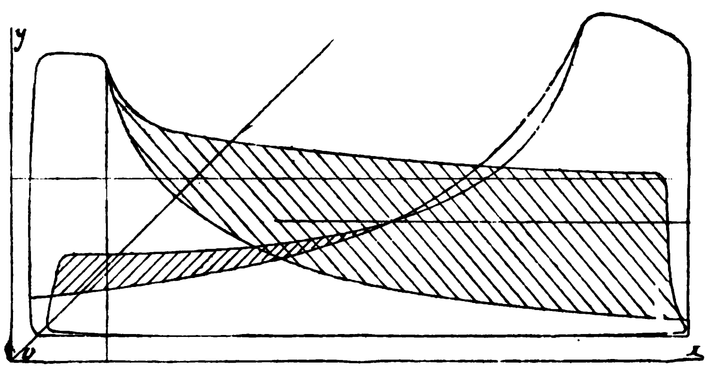


FIG. 102.

is in the form of a hyperbola, which as a general principle may be taken to be fairly correct.

After the steam is cut off it still condenses in the cylinder if the walls are not hot enough and the curve follows more or less the form of a hyperbola, but soon, owing to partial re-evaporation, the curve rises and continues at a higher level than that of Mari-

otte to the end of the stroke. The diagrams (Fig. 102) taken on the engines of the *Michigan*, at different speeds, show very clearly the value of this re-evaporation. Sometimes the actual curve rises apparently from the very commencement of the cut-off.

At first, as Mariotte's law was considered to give the exact expression for the expansion of steam, this rise in the diagram was attributed to leakage of high pressure steam into the cylinder; but as no accuracy in the construction of the valves made any difference, it was gradually recognised that some of the water condensed on the admission of steam becomes re-evaporated during the fall of temperature under expansion, owing to its own latent heat and the heat transmitted to the cylinder walls during the admission of steam. Re-evaporation ceases before all the condensed water has been turned to steam again, and the curve falls in consequence, showing a fall in pressure. Re-evaporation is never complete. A certain amount of water remains in the cylinder, and being driven back by the return stroke whilst the steam exhausts, tends to decrease the heat of the steam admitted at the next stroke.

In the case of locomotive engines, the curve of the forward end of the cylinder is always lower than that of the back end, owing to the cooling of the cylinder end, which meets the wind. Only by study of the diagram can one fix upon the most economical cut-off for any given engine, namely that which shows at the end of the stroke the minimum pressure which is able to overcome the friction of the engine.

Expansion in the Case of Two-Cylinder Engines.

The work given out by steam is the same for two cylinders as for one, but it is more economical to use two, for then the high pressure cylinder is not in direct connection with either the outer air or the condenser, and high pressure steam does not pass direct into the second cylinder; the extremes of temperature are less, therefore, in each of the two cylinders than in the one, and the amount of steam condensed on admission and during expansion is less.

This is more noticeable in a compound engine fitted with a steam reservoir. In the case of an ordinary compound engine, there is pressure of steam on both pistons at the end of the stroke; whereas in the case of the compound engine fitted with a steam reservoir, the steam exhausts into this reservoir from the high pressure cylinder and is re-admitted to the low pressure cylinder again as though direct from a boiler; with the result that, assuming a constant pressure in the reservoir, the back pressure on the exhaust steam from the high pressure cylinder will be constant, so that all things being in other respects equal, the difference of pressure, and therefore extremes of temperature and consequent condensation in the high pressure cylinder, is, where a steam reservoir is used, less than in the case of a high and low pressure cylinder directly connected with one another.

Steam Jacketing.

The steam jacket was invented by James Watt. Its efficiency has often been questioned, but the best engine builders have recognised its value.

The steam jacket which encloses the whole cylinder has the effect of keeping the internal walls of the cylinder as nearly as possible at the same temperature as the steam admitted to the cylinder; condensation is therefore reduced to a minimum and the curve closely follows that of Mariotte.

The jacket is only kept hot in effect by the condensation, within the jacket, of a certain amount of

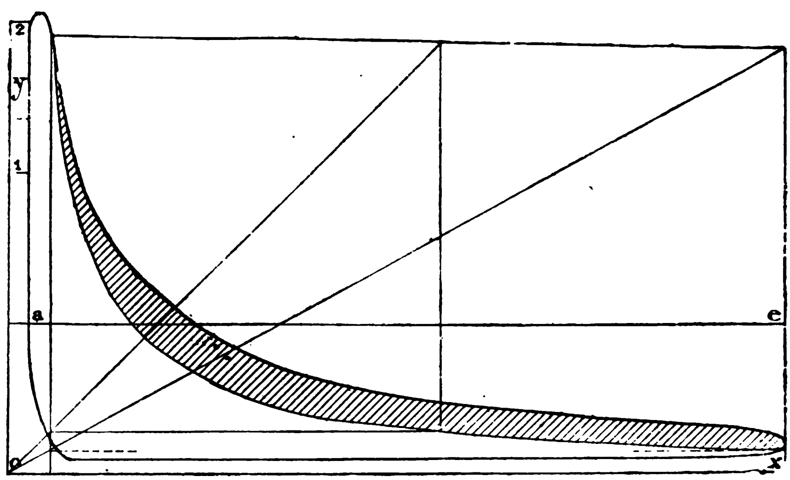


FIG. 103.

steam. This water must constantly be discharged by means of blow-off cocks, or better still, pumped back into the boiler. This water does not re-evaporate and does not therefore rob the walls of the cylinder of any other heat. Care must be taken that none of the water from the jacket can get into the cylinder through the steam pipes.

The greater the extremes of temperature of the steam the more important it is to steam jacket the engine. Steam jacketing is of great importance in

the case of condensing engines, and in cases where there is a wide range of expansion within one cylinder.

It is of less importance in the case of two cylinder engines with little expansion—especially as regards the high pressure cylinder. Steam should be able to circulate in the jacket. Air will be driven out of the jacket equally well by blow-off cocks below the cylinder, as the density of air is at all temperatures greater than that of steam; but all air must be driven out in order that the hot steam may everywhere be in direct contact with the cylinder.

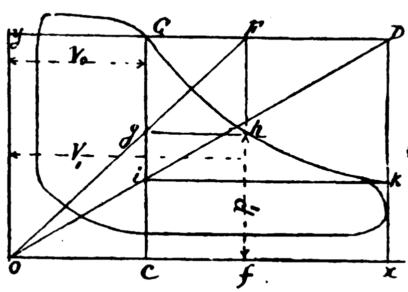


FIG. 104.

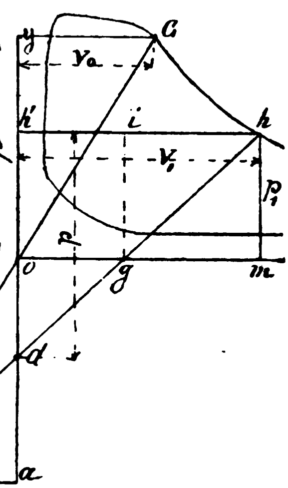


FIG. 105.

Mariotte's Curve (Fig. 104).

Mariotte's law,[1] which states that the pressure of steam varies *in inverse ratio to its volume*, may be graphically represented by a hyperbola.

This law states that: $P\ V=$constant.

[1] Generally called Boyle's law in this country.

Let V' represent waste spaces, then C being the space traversed by the piston, $C+V'$ is the volume at the end of the stroke (V). V_0 (Fig. 96), the volume of steam on admission, includes V'.

From the above, it follows that, if we take the abscissae as representing volumes equal to 2, 3 and 4 times V_0, the corresponding ordinates will be equal to $\frac{1}{2}$, $\frac{1}{3}$ and $\frac{1}{4}$ the initial pressure p at the commencement of expansion—and in this way as many points as desired may be found and a curve drawn.

Diagrammatic Representation.

From C (Fig. 104) representing the pressure $C\ c=p$ at the commencement of expansion, draw the horizontal line $C\ D$ parallel to $o\ x$. If, at a point F, corresponding to volume V, we rule a vertical line $F\ f$ and a diagonal $F\ o$, this latter cuts $C\ c$ at a point g, which, carried to join $F\ f$ at h, gives a position on the curve, for we have

$$\frac{g\ c}{F\ f}=\frac{o\ c}{o\ f} \text{ or } \frac{p_{\prime}}{p}=\frac{V_0}{V_{\prime}}$$

The extreme point of the curve K may be found in the same way:

$$K\ x=i\ c=p_{\prime} \text{ and } \frac{i\ c}{D\ x}=\frac{V_0}{V}=\frac{p_0}{p}$$

In this way any number of points on the curve may be determined. Again, if C (Fig. 105), the point at which the original volume V_0 begins to expand, has a pressure p, draw the line $o\ a=o\ y=p$ and $a\ b=V_0$. The diagonal $C\ b$ passes through o and forms two similar right-angled triangles. From b draw a line $b\ h$,

cutting $o\ a$ at d and $o\ x$ at g. Make $g\ h = b\ d$ and h will be a point on the hyperbola. Moreover draw $h\ h'$ parallel to $o\ x$ and $g\ i$ and $h\ m$ perpendicular to $o\ x$.

Clearly $h'\ d = p$ and $\frac{i\ g}{h'\ d} = \frac{i\ h}{h\ h'}$ or $\frac{p'}{p} = \frac{V_\circ}{V'}$.

Advance of Exhaust.

This should increase in proportion as the final steam pressure is high and the speed of the engine considerable, in order to insure the depression at the return stroke and the compression on the other side of the piston.

In Fig. 106 the shaded parts of A, B, and C represent

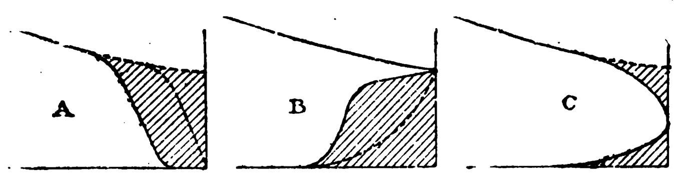


FIG. 106.

loss of work. In the case of A the advance is too great —in that of B the exhaust is actually delayed, and the dotted line shows what the diagram would be if there were no advance of exhaust. C shows the least loss, and therefore the best degree of advance exhaust.

The expansion curve often continues without showing any appreciable fall in pressure, even after the commencement of the exhaust, owing to the rapid re-evaporation of the water carried in with the steam, which may be suspended in minute particles or lie on the walls of the cylinder.

Exhaust and Back Pressure.

At the commencement of the back stroke of the piston the pressure falls in proportion to the condensation, or the extent to which the steam is free to exhaust into the open air. This back pressure is greater when the escape ports are not of sufficient area or when the exhaust steam is employed to heat the feed water.

At the commencement of exhaust, the water which is held in suspension in the steam, or lies on the walls of the cylinder, evaporates rapidly, especially in the case of condensation, as it takes up the heat from the walls of the cylinder. The pressure is therefore higher than it would be if it were merely the pressure to which the steam had fallen by expansion.

But if the temperature of the water of the cylinder has already fallen owing to a long period of expansion and small admission of high pressure steam, all the water cannot re-evaporate, and a certain quantity accumulates in the cylinder at each stroke.

This is the cause of the knocking which happens in many cylinders, and which can only be got rid of by keeping the blow-off cocks constantly open, and is one reason why it is a good thing to have the exhaust valve chamber underneath the cylinder. It also shows the disadvantage of having much expansion in the case of locomotive engines. In the case of condensing engines, the back pressure is from 0·10 to 0·15 of an atmosphere—equal to 3 to 4·5 inches of mercury, whilst in non-condensing engines it reaches from 1·10 to 1·20 of an atmosphere.

Back pressure is greater when the steam is wet. According to Clark, with constant piston speeds it is proportional to the pressures at the end of expansion, and with constant pressures it is proportional to the squares of the speed of the piston.

Compression.

As soon as the exhaust ports close (Fig. 107) the steam which remains in the cylinder is compressed

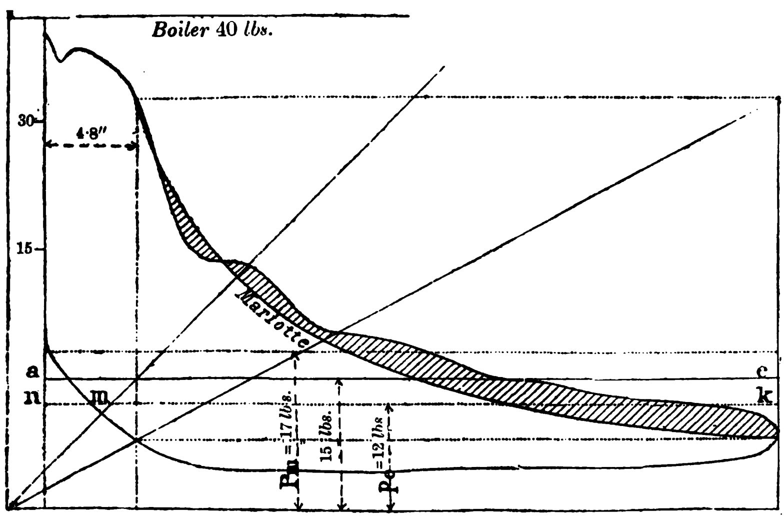


FIG. 107.

by the piston; its pressure rises, following a regular law as in expansion, owing to the exchange of heat between the steam which becomes heated by compression and the water and walls of the cylinder, and the exact reverse takes place to what happens under expansion. It seems, therefore, that the curve of the diagram should lie below that of Mariotte.

The work taken up during compression is given out

again on the return stroke either in power or in heat.

For any given horse power, therefore, the diameter of the piston should be a little larger if there is compression than if there is none. But the advantages of compression outweigh this disadvantage.

In the first place, the temperature of the steam, rising under compression, causes the water in suspension and that on the walls of the cylinder to re-evaporate, and so decreases the condensation on the next admission of steam.

Secondly, the steam compressed in the neutral space beyond the travel of the piston actually effects an economy in the expenditure of steam, as the incoming steam has not to fill this amount of space, but only that through which the piston travels.

Thirdly, the gradual resistance offered to the piston by this compression forms a buffer, and by keeping the moving parts of the connecting rod, etc., always pressed together, counteracts their natural inertia, and lessens the shock at the change of stroke from one direction to another.

With compression the advance admission may be reduced. The larger the amount of waste space in the cylinder, the sooner should compression begin. This is adjusted by the slide valve. It is difficult to determine what degree of compression is the most economical. If it is admitted that the work given out in compression is equally restored during the return stroke or in heat, the highest economy will be effected when the pressure of the compressed steam is equal to the pressure of the fresh steam from the boiler.

Let V' be the waste space and p the compression at the end of the stroke; V_0 the volume of steam and p' the amount of compression at the moment when compression begins; then according to Mariotte's law

$$p = p' \frac{V_0 + V'}{V'}$$

If p be greater than p' the slide valve will be lifted off its face, and to prevent this the clearance space must be increased. The most economical amount of compression can be judged better from a diagram than from any number of calculations.

The Compression Curve.

Accepting Mariotte's law and knowing the point a (Fig. 108) where compression begins, draw the horizontal line $a\ b$ and the vertical line $a\ c$.

If from any point m on $a\ c$ we draw the horizontal line $m\ n$ and the diagonal $m\ o$, the last will cut $a\ b$ at a point d, a vertical line through which will cut $m\ n$ in the point e, which lies on the required curve; and the bisector of the angle $y\ o\ x$ is the axis of the hyperbola on which all the points like e lie.

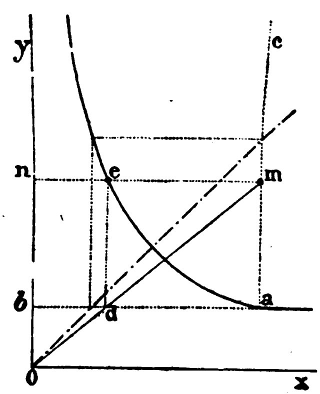


FIG. 108.

To find the Amount of the Clearance Space.

Draw any line, cutting the curve (Fig. 109) at two points $a\ b$, and suppose that it cuts $o\ x$ at c.

If we make $b\ d = a\ c$, the point d will fix the position of the ordinate $o\ y$, the distance of which from the diagram $n\ b = m\ c = V'$ the waste space.

and $\frac{a\ m}{b\ q} = \frac{m\ c}{q\ c}$ or $\frac{p'}{p} = \frac{m\ c}{q\ c}$

but the right-angled triangles $d\ n\ b$ and $a\ m\ c$ are

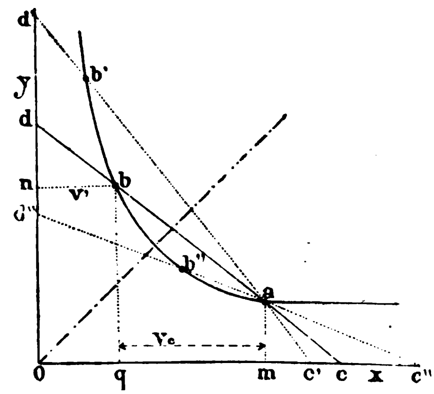


FIG. 109.

equal, therefore $n\ b = m\ c = V'$ the volume after compression, and $q\ c = o\ m = V$ the volume before compression; therefore $\frac{p'}{p} = \frac{V'}{V}$

From this we arrive at another method of tracing the curve. When we know $o\ x$, $o\ y$ and the point a where compression begins, we have only then to rule oblique lines through a, and, starting from $o\ y$, make them the following lengths: $d\ b = a\ c$, $d'\ b' = a\ c'$, $d''\ b'' = a\ c''$, etc.

Expansion in Compound Two Cylinder Engines.

The amount of horse power given out in these engines is the same as in the single cylinder engine,

but the economy in steam is greater. As the high pressure cylinder is neither open to the air nor to the condenser, and as steam cannot enter the low pressure cylinder direct from the boiler, the extreme differences of temperature in the cylinders is less, and consequently condensation on admission and expansion is reduced to a minimum.

Diagrams taken from a Compound Engine (Figs. 110–112).

These are taken from the high pressure cylinder and the low pressure cylinder of a compound engine with a steam receiver between the cylinders—the indicator being fitted with different and suitable springs.

In order to compare the expansion with that which would take place in a single cylinder, following Mariotte's law, the volume and steam pressure in each must be reduced to the same scale.

The important dimensions in this case are as follows:

Diameter of high pressure piston	= 11·2	inches.
„ low „ „	= 18·9	„
„ piston rod in both cylinders	= 2·2	„
Travel of each piston	= 18·9	„
Ratio of high to low pressure cylinder	1 to 2·84	

The base lines of the diagrams (Figs. 110 and 111) are divided into ten equal parts.

Draw the line of volumes $O\,X$ (Fig. 112) and the line of pressures $O\,Y$. From O mark off distances to represent to scale the volumes of the low and high

pressure cylinders, and on these new bases replot the two diagrams (Figs. 110, 111), using the same scale of pressures for both curves.

Then trace Mariotte's curve, starting from the point

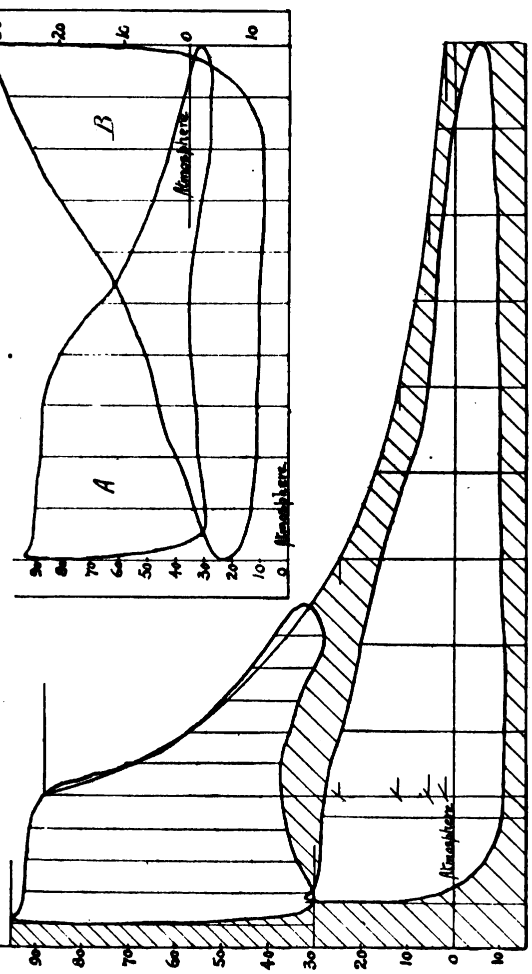


FIG. 111.

FIG. 110.

FIG. 112.

where expansion commences in the high pressure cylinder.

The piece of the high pressure diagram which over-

laps Mariotte's curve represents the work done by re-evaporation. The fall of pressure between the two cylinders is found by measuring the difference between the ordinates at the half exhaust of the H.p. cylinder and the admission to the L.p., the crank being at 90 degrees.

Diagram taken for a Compound Engine without Intermediate Steam Receiver.

If the diagrams of a compound engine without intermediate steam receiver are reduced to one scale of

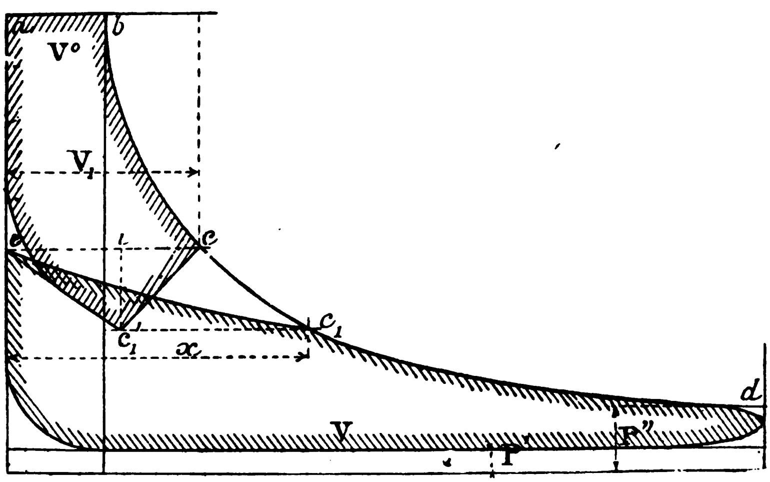


FIG. 113.

volume and pressure as in the preceding case, the diagrams obtained are as shown in Fig. 113.

Diagram taken with a Triple Expansion Engine.

Proceeding as in the above case, we get the diagram Fig. 114.

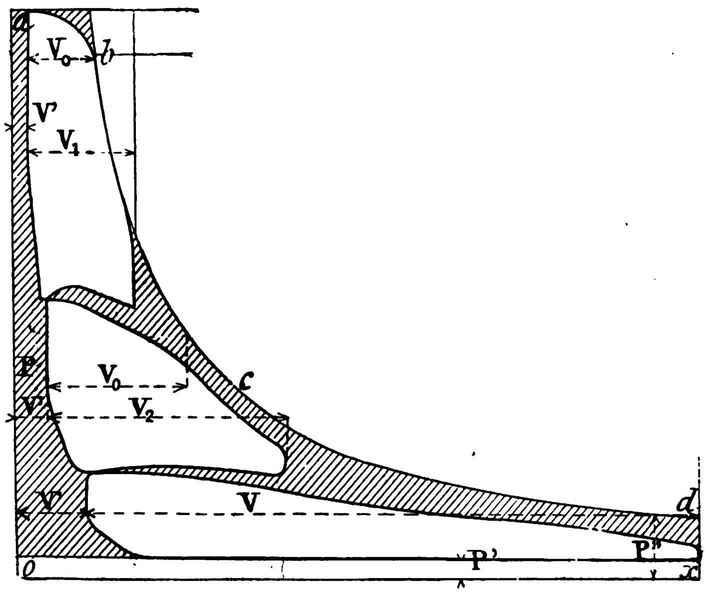


FIG. 114.

CAUSES OF ERRORS IN DIAGRAMS.

One cause of error arises from the paper drum; for instance, if the tracings are not in true proportion to the piston travel, or if the impulse imparted to it causes an elongated diagram.

Another fact to be remembered is that it is essential to take diagrams at each side of the piston. On one side the diagram is sure to show better results than on the other.

Duplicates of diagrams, traced by hand, should never be trusted.

The exact position of the atmospheric line can only be made sure of when it has been traced by the indicator itself.

Various Applications of the Indicator.

To Fix the Point of Admission.

Mr. Graham has employed the indicator to trace diagrams of which the ordinates are proportional to the steam pressure, but of which the abscissae are proportional to the travel of the slide valve.

Having taken the diagram $A\ C$ (Fig. 115) on one

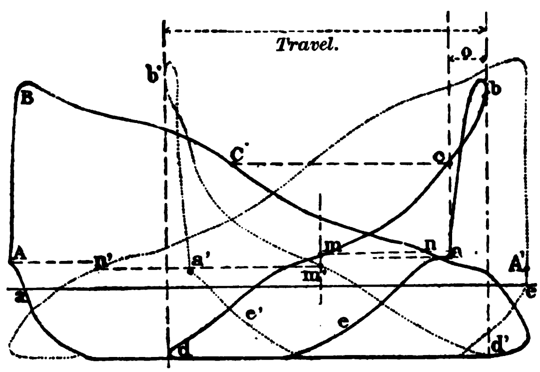


FIG. 115.

side of the cylinder, the drum was connected to the slide valve rod, and the diagram $a\ b\ c\ d\ e$ was obtained.

Point a is where admission begins, and corresponds to A. Pressure rises up to b at the same height as B, then on the return travel of the valve it falls as the ports close. Point c is on a vertical line drawn from a, where the ports close; it fixes definitely C as the point where expansion begins.

The length $d'd = s$ represents, on a definite scale, the travel S of the valve, and O on the same scale the maximum amount of opening of the port. The actual opening of the port to admit steam is then $\frac{O}{s} \times S$.

The dotted diagrams are similar ones taken on the other side of the piston.

The distance $a\ a'$ is equal to double the external lap supposed to be equal on each side. The vertical line drawn midway between a and a' will give the points m and m', which, if there is no internal lap, will determine at once the points n and n' which mark the commencement of the exhaust.

If the valve has internal laps r and r', and external laps R and R', we can determine the points m and m' by drawing a vertical line distant from a' by a length $R + r$, and another vertical line distant from a' by the length $R' + r'$.

The indicator may be fixed on any portion of the steam system where the pressure varies during the stroke.

1. *On the boiler* if the position of the engine allows of it. In this case the indicator drum is worked by a brass wire. The changes in pressure in the boiler, at each stroke of the engine, can be noted. When these variations are considerable the result frequently is that water is forced into the steam pipes. Such variations in pressure are generally found in the cases of boilers whose steam-raising capacity is too small for the requirements of the engine which they feed.

2. *On the Steam Chest.*—In comparing the pressures shown by the indicator, first at the boiler and then at the cylinder during admission of steam, the fall of pressure caused by the flow through the steam pipes and the valves into the cylinder can be determined.

Diagrams 1, 2, 3, 4 (Fig. 116), borrowed from Mr. Porter's work, were taken from the steam chest of an

Allen engine running at a speed of 200 revolutions per minute.

From the moment when the ports open at the dead points *a*, the pressure falls more or less to *b*, where they close. The pressure then rises owing to the inertia of the moving column of steam, and falls again towards *a*. In diagram 2 the admission is shorter, and

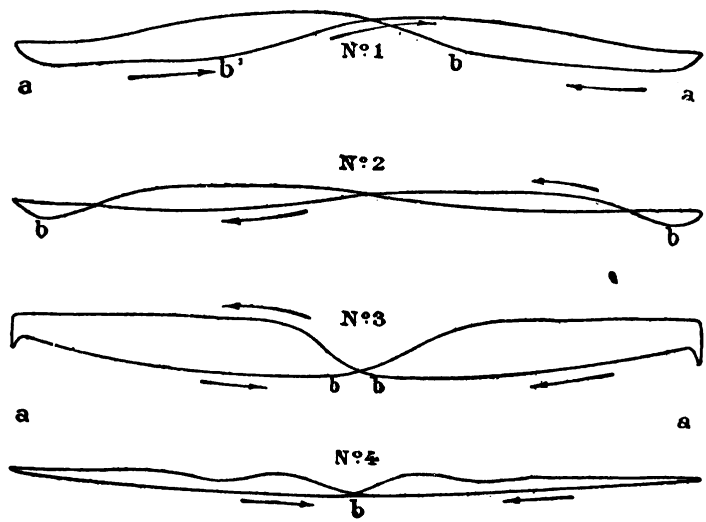


FIG. 116.

the pressure rises less than in the former case after the closing at *b*. The reverse is the case in diagram 3: the pressure falls from *a* to *b* in proportion as the speed of the engine increases.

At *b* the flow of steam is most rapid, and the pressure rises rapidly. The result of this is a marked fall in pressure at *a*.

The variations in pressure resulting from the inertia of the column of steam increase with the revolutions

of the engine and the speed at which the valves close.

If the steam pipes from the boiler are long, or if a pressure regulator is used between the boiler and the engine, it is advisable, in order to lessen the loss of pressure, to fix a steam reservoir close to the engine of an area equal to that of the cylinder itself. Diagram 4 shows how the changes in pressure are lessened by the use of a steam reservoir.

3. *On the Exhaust Pipe.*—The indicator shows, especially in the case of non-condensing engines, how much of the back pressure is due to cushioning and how much to the exhaust pipe by comparing the pressures obtained with those inside the cylinder.

4. *On the Condenser and on the Air Pump.*—The indicator shows the amount of vacuum in the one case and the horse power used in the other.

The Indicator Applied to Hydraulic Machinery.

The indicator has been applied to pumps, to high pressure water pipes serving hydraulic lifts, etc., and to hydraulic riveting machines.

We can only here indicate the modifications necessary for very high pressures.

The springs for these high pressures are difficult to make, and it is preferable to proportion the area of the piston to the pressure which it is desired to measure, whilst using any springs one may select.

In the indicator shown in Fig. 117 (made by Dreyer, Rosenkranz & Droop) the lower portion, *P*, forms the cylinder of reduced area, the plunger *K* and the spring

are mounted on a specially constructed rod. In some cases the small cylinder E is used as shown.

If four springs are used to cover a range of pressure of from 2 to 15 atmospheres on a plunger of 0·8 inch, the pressures which can be measured will vary inversely

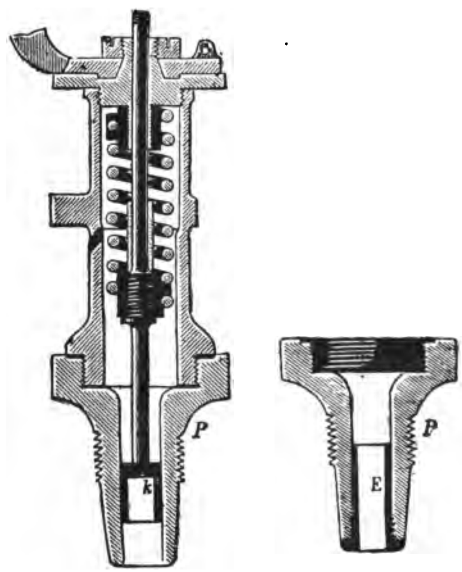


FIG. 117.

with the areas. The following table gives these pressures in the case of two pistons of 0·5 inch and 0·35 inch diameter:—

Diameter of pistons in inches. Ratio of Areas.	0·8 1	0·5 $\frac{2}{5}$	0·35 $\frac{1}{6}$
Limit of pressures in lbs. per square inch.	30 75 120 225	75 187·5 300 555	180 450 720 1350

CHAPTER VI.

THE TESTING OF GAS AND OIL ENGINES.

A SIMILAR diagram to that given by the steam engine may be taken from a gas engine, the diagram showing the force of the explosion, varying as it does with the degree of air in combination with the gas, petroleum, benzoline, etc., and the proper amount of compression. The area of the diagram is the measure of the horse power for any given cycle.

The diagrams (Fig. 118) are taken from an engine

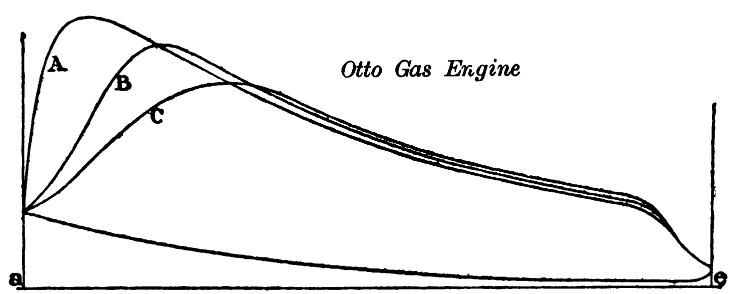


FIG. 118.

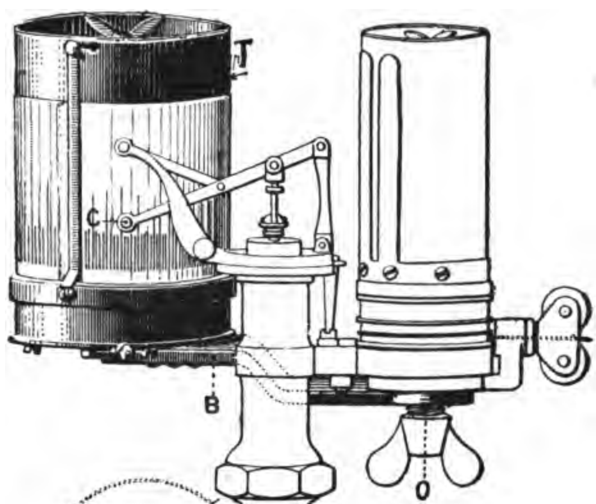
using pure gas. The curve *A* is taken with the minimum amount of air requisite to give the best result.

B and *C* show successively larger proportions of air.

Mathot's Registering Indicator, made by P. Garnier.

The diagram gives one cycle only, which is not

Fig. 119.



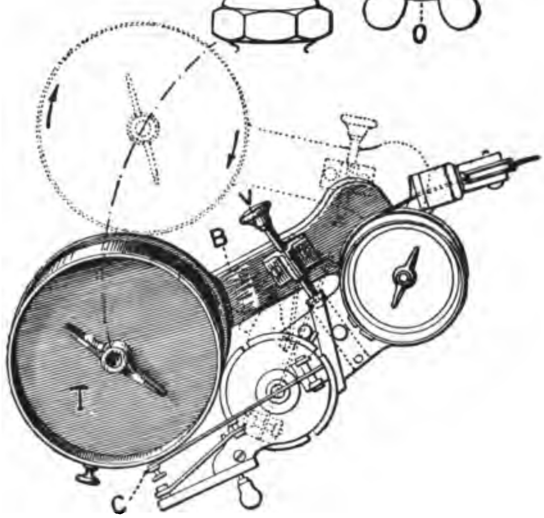


Fig. 120.

enough to give a correct record of the work of a gas engine, as successive explosions differ from one

another. Mathot has devised the following way of getting over this difficulty:—

To the ordinary indicator (Figs. 119 and 120) is added a second paper drum T carried on B, which is clamped to the base of the ordinary drum by the hand-tightened nut O. This drum T is actuated by clockwork, the speed of which is controlled by a special regulator. At each complete révolution of T the paper must be renewed.

In order to obtain a diagram showing successive explosions or the ordinary closed diagram, the drum T or the ordinary drum may be brought into position under the pencil.

In the case of the continuous registering indicator, the paper, which is in one long strip rolled up within a drum is drawn off into a second drum by the action of clockwork.

Conditions necessary for Testing. Analysis of Diagrams.

With regard to the working of these indicators we cannot, in our opinion, do better than quote the words of the inventor. There are many considerations to be taken into account, and the method employed varies in respect to which particular phase of the cycle it is desired to study.

I.—*To find the Amount of Compression.*

Use a medium spring, of which the full play corresponds approximately to the maximum amount of compression, in order to obtain a curve on as large a scale as possible.

The usual practice in the testing room is to drive

the engine, light, by means of an electric motor, at varying speeds.

The compression of the mixture of gas and air in the cylinder decreases as the number of revolutions of the shaft increases, owing to the resistance set up in pipes and valves, which resistance increases in proportion to the speed.

Fig. 121 shows portions of two tracings, taken in different tests.

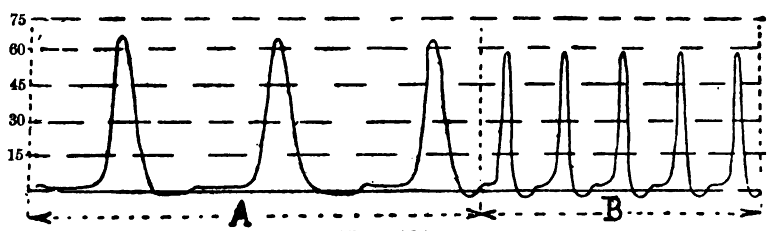


FIG. 121.

A. Speed of engine, 950 revolutions; compression, 71 lbs. per square inch.

B. Speed of engine, 1,500 revolutions; compression, 63 lbs. per square inch.

Hence there is a drop of pressure of 11·5 per cent.

II.—*To measure the Resistances to Admission and Exhaust* (Figs. 122, 123).

We will show first the effect of the tension of the admission valve spring and of the area of the valve; and then the effects of the area of the exhaust valve and of the length and shape of the exhaust pipe.

Use a very light spring, the extent of play of which can be regulated by a pin, so as to obtain, on a relatively large scale, the depressions and resistances

which are respectively indicated by the curve corresponding to them, whether below or above the atmospheric line.

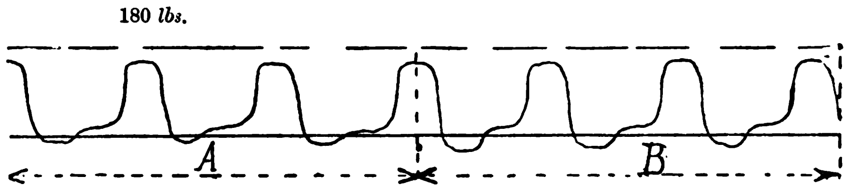


FIG. 122.

A. Fig. 122. Tension of the inlet valve, 2·09 lbs.; resistance to intake, $\frac{1}{7}$ atmosphere.

B. Fig. 122. Tension of the inlet valve, 4·18 lbs.; resistance to intake, $\frac{2}{7}$ atmosphere.

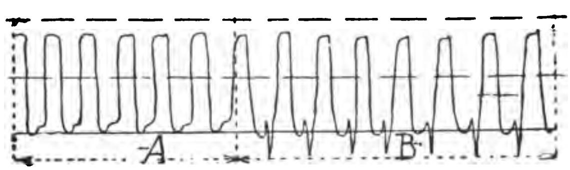


FIG. 123.

A. Fig. 123. Exhaust into a trap; resistance to exhaust, $\frac{2}{7}$ atmosphere.

B. Fig. 123. Exhaust into open air; the exhaust pipe and trap being removed, the resistance to exhaust is nil.

One may assume that the amount of depression shown by the tracing is partly due to the inertia of the spring and the registering apparatus, as the spring is slackened abruptly on the opening of the exhaust valve.

III.—*To Compare the Mean Pressures of Explosions by means of Ordinates in Juxtaposition* (Fig. 124).

Use a strong spring, and adjust the speed of the paper drum so as to obtain ordinates corresponding as nearly as possible to the explosions.

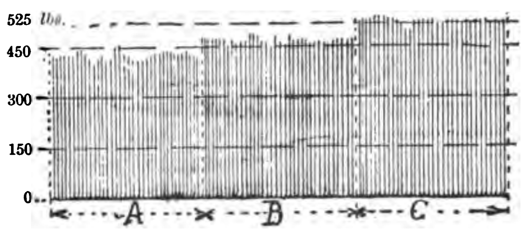


FIG. 124.

A. Fig. 124. Pure alcohol, explosive force from 400 to 450 lbs. per square inch.

B. Fig. 124. Carburetted alcohol (*electrine*), explosive force from 450 to 480 lbs. per square inch.

C. Fig. 124. Essence of petrol (*stelline*), explosive force of from 480 to 530 lbs. per square inch.

IV.—*Analysis of the Cycle by means of open Diagrams representing Four Impulses.*

The speed of the engine from which the diagrams in Fig. 125 were taken was 1,200 revolutions per

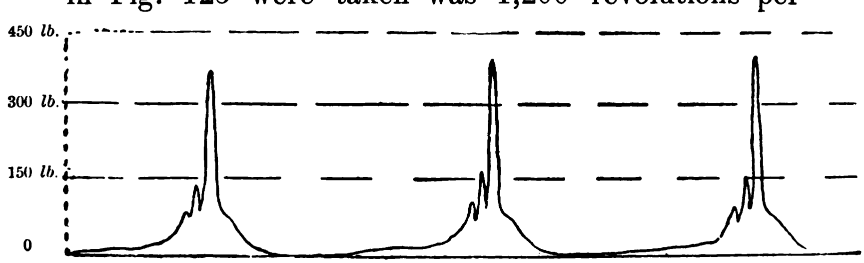


FIG. 125.

minute; carburetted alcohol was used; maximum pressure of explosions, 425 lbs.; mean compression, 64 lbs.; pressure at the end of expansion, 21·5 lbs., etc.

Use a strong spring and let the paper move quickly over the drum. The four phases of the cycle will appear clearly, traced one after the other from right to left; that is to say, in reverse order to the unwinding of the paper, tracing an open diagram exactly showing the pressures at different stages of the piston travel.

The tracing of the phases of the cycle is as true a record as if obtained by means of an indicator registering a closed curve.

So far as concerns the reading of the diagrams, it does not matter if the movement of the paper is not rigidly in step with that of the piston of the engine.

Efforts have been made to obtain open diagrams by means of registering apparatus in which the movement of the paper is actuated by the engine itself, but neither such apparatus, nor ordinary indicators, are suitable when the speed of the engine exceeds 400 revolutions per minute.

V.—*Analysis of the Effects of Inertia in the Indicator. Choice of Spring* (Fig. 126).

Knowing the speed with which the explosions follow one another in engines used for propelling motor cars, it is evident that the effect of inertia of the various parts of the indicator must be visible on the diagram.

The amount of these irregularities depends upon

the weight of the moving parts and the area covered by them.

The moving parts are the plunger and its rod, the spring and the lower arms of the parallelogram.

The effect of inertia has been reduced to a minimum by keeping the weight down. The plunger is grooved out, all needless metal being cut away. The plunger rod is hollow and the lever arms are short and light. A

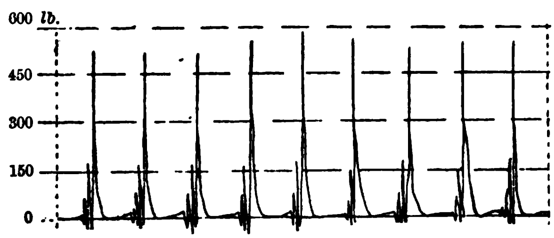


FIG. 126.

silver point, working on prepared paper, takes the place of a lead pencil, and light springs with small play are used.

The interior of the cylinder above the plunger must be well oiled each time the spring is changed, and this oil at each throw of the plunger is splashed over the walls of the cylinder.

If from want of precaution—particularly that of selecting a suitable spring—effects of inertia are produced, they are easily detected on the diagram, and need not be confounded with the curves representing the phenomena in the engine cylinder.

The cylinder of the indicator is kept cool by water circulating in a jacket.

As the explosion chamber in the case of motor car

engines is very small, care must be taken not to sensibly increase it, or the working conditions will be altered. With this object in view, the indicator cylinder is so arranged that the plunger is no higher than the cock which shuts it off from the cylinder; that is to say, there is no length of pipe leading to it.

Diagrams taken with certain optical apparatus, in which the gases fill a long pipe of small section, show very considerable distortions.

CHAPTER VII.

MEASURE OF INDICATED H.P.

Measurement of Mean Pressure—Trapezium Method.

IF the line of the diagram undulates, a direct line must be traced, passing through the middle of the undulations. This line is more true than the line of equal surface.

If several curves have been traced, one over the other, a mean of them must be traced by hand.

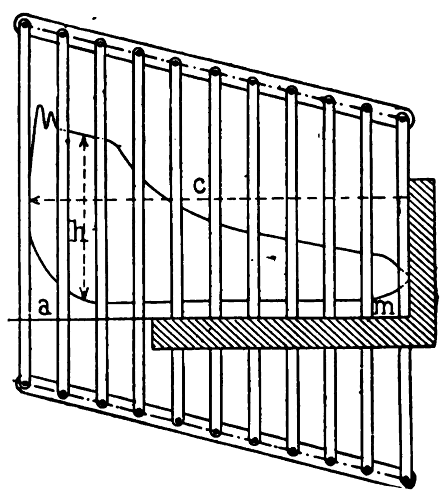


FIG. 127.

This done, divide the diagram into ten equal sections by means of an apparatus (Fig. 127) con-

sisting of parallel equidistant rules, rising from a perpendicular base line, fixed by a set square; then trace the mean height of each trapezium. The mean ordinate y_m is the arithmetical mean height of these ten sections.

Graham's screw (Fig. 128) gives y_m directly.

It is constructed as follows:—a screw f tapped into the screw-nut e, carries at one end a disc whose circumference is equal to ten threads of the screw. Let e remain held in the hand without revolving, and trace over the ordinates of the diagram with the pointer d, the disc revolving as it rolls over the paper; then it is clear that the distance a b will represent the mean ordinate y_m.

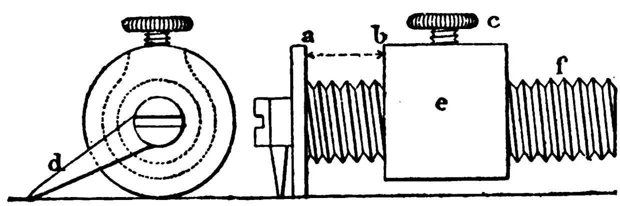


FIG. 128.

Again, the sum of the ordinates may be measured with a pair of compasses or a slip of paper.

Having divided the diagram, measure the ordinates with a scale equal to ten times the scale e of the spring; then add the various lengths together, and the sum is the mean pressure. In symbols,

$$\frac{10\Sigma y}{10e} = \frac{\Sigma y}{e} = p_m$$

Another Method.

In the case of very irregular curves, each trapezium

must be divided in 2, 3 or 4 equal vertical sections, and the mean heights taken, in order to arrive at the value of the area of the first trapezium. In Fig. 129

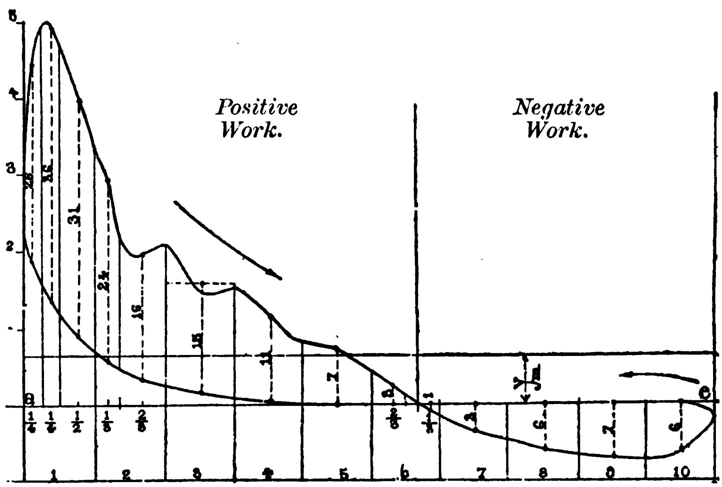


FIG. 129.

the first trapezium is divided into two quarters and one half, the second and sixth are divided into one-third and two-thirds—then proceed as follows:—

No. 1,	$\frac{1}{4}$ (26 + 36)	= 15·5 tenths of an inch.
	$\frac{1}{2}$ (31)	= 15·5
„ 2,	$\frac{1}{3}$ (24)	= 8·0
	$\frac{2}{3}$ (16)	= 10·7
„ 3,		= 15·0
„ 4,		= 11·0
„ 5,		= 7·0
„ 6,	$\frac{2}{3}$ (3)	= 2·0

Sum of positive ordinates = 84·7

No. 6, $\frac{1}{3}$ (1)	= 0·3
„ 7,	= 3
„ 8,	= 6
„ 9,	= 7
„ 10,	= 6

Sum of negative ordinates = – 22·3.

Therefore total sum = 62·4.

And therefore mean height = 6·24 tenths of an inch.

The resultant positive difference (62·4) between the totals of the positive and negative ordinates must be divided by 10 (the number of divisions) in order to ascertain the mean ordinate of the diagram.

Simpson's Method.

Divide up the diagram (Fig. 130) into an even num-

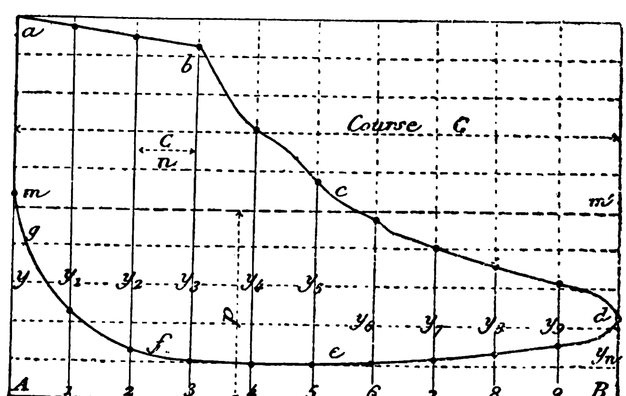


FIG. 130.

ber n of equal spaces (the size of these being determined by the degree of irregularity of the curves and the degree of exactitude required). Measure the ordinates $y, y_1, y_2, \ldots \ldots y_n$.

The mean ordinate is—

$$y_m = \frac{1}{3n}\left[y + y_n + 2\,(y_2 + y_4 + \ldots + y_{n-2}) + 4\,(y_1 + y_3 + \ldots + y_{n-1})\right]$$

Certain of the ordinates may, as in the case of y_n be zero, but this does not affect the formula.

Remarks.

Whatever the method employed, the result is more exact, although it takes longer to obtain, if the mean ordinate y'_m of the upper curve and that y''_m of the lower curve down to the line of absolute vacuum, be

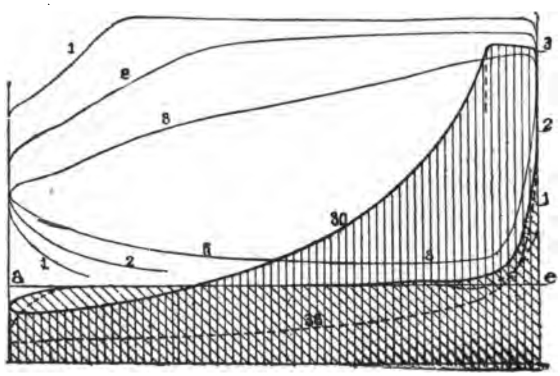


FIG. 131.

taken separately, and the mean pressure found by the formula $y_m = y'_m - y''_m$.

If y'_m and y''_m have been measured from the atmospheric line, $y_m = y'_m - y''_m$ for non-condensing engines and $y_m = y'_m + y''_m$ for condensing engines.

The first method—namely, taking the vacuum line as axis—is the more general, and there is less chance of error in confusing the positive and negative work when the diagram forms a loop. In the diagram Fig. 131

the vertical shading shows the area representing the positive work, and the oblique shading that representing the negative work, the vacuum line being taken as axis.

FIG. 132.

Amsler's Planimeter
(Figs. 132, 133).

This planimeter was invented in 1855 by Professor Amsler of Schaffhausen. As its name indicates, it measures the areas of surfaces in one plane. By a slight modification it can be used to measure the mean height of a diagram. If the scale of the spring be different for extension and compression, we must measure separately the mean ordinate of that part of the diagram above the atmospheric line and the mean ordinate of the part below it.

The paper upon which the diagram is traced is placed on a smooth flat surface. The distance apart of the points O and O (Fig. 132) are adjusted by moving the cursor along the bar until O O is nearly equal to

the breadth of the diagram. The cursor is then clamped, and the final adjustment of the distance beween O and O is made by means of the tangent screw

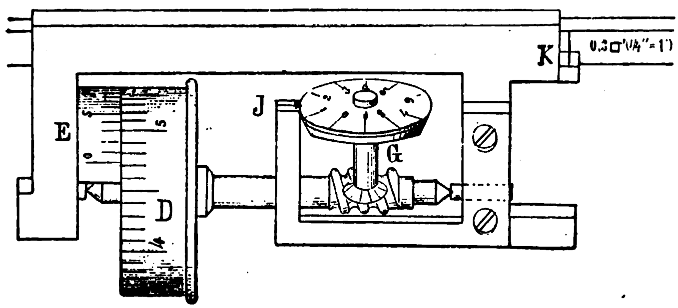


FIG. 133.

M. The zero of the wheel G is placed opposite its index J (Fig. 133), and the zero of the rolling wheel

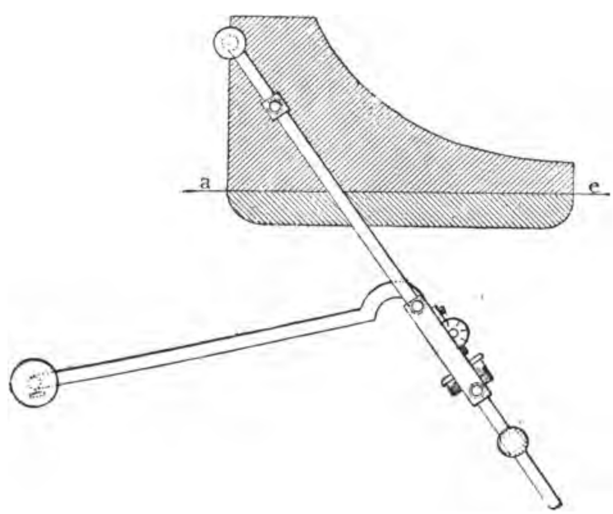


FIG. 134.

D is placed accurately opposite the zero of the vernier E.

The instrument is then placed on the plane in the

position shown in Fig. 134, and the needle point E of the movable arm is lightly pressed on the paper and is kept in its position by a weight placed on it. The pointer F is taken gently round the curve of the diagram in the same direction as the pencil of the indicator went. During this operation the rim of the rolling wheel D must constantly bear on the paper and must not move backwards or forwards over its edges. The point E therefore has to be carefully chosen. The pointer having made a complete circuit of the diagram, we read the numbers on the wheel G, the rolling wheel D and the vernier E (Fig. 133).

Suppose, for example, that the index J is between the 1 and 2 marked on wheel G, that the zero of the vernier points between 47 and 48 on the wheel D, and that the vernier reading is 3. Then the reading of the instrument is 147·3 and the mean ordinate is

$$\frac{147 \cdot 3}{20} = 7 \cdot 365.$$

This may be in inches or centimetres depending on how the instrument is graduated. The number 20 is arranged by the maker's graduation of the instrument.

Since the pointer F follows the trace made by the pencil of the indicator, no error results if the diagram is looped, as the reading of the instrument is proportional to the difference between the areas of the two portions of the diagram, and hence no correction has to be applied to the reading in this case.

Theory of the Planimeter.

Consider the rolling wheel P in Fig. 135. Let its axis $a\,c$ make an angle α with the line $a\,b$ and make it traverse the line always keeping its axis parallel to its

original direction. When it arrives at b the wheel will have turned through exactly the same angle as if it had travelled first along $a\ c$ and then along $c\ b$. From a to c its rotation would be zero, and from c to b there would be pure rotation with no slipping; $c\ b$ may therefore be taken as a measure of the rotation. Also

$$c\ b = a\ b \sin \alpha.$$

Suppose now that when the rolling wheel comes to b its axis is turned round until it makes an angle β

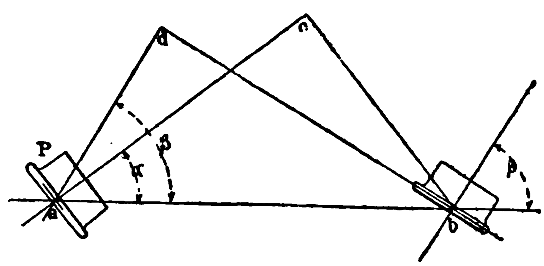


FIG. 135.

with $a\ b$. Now let it move back to a, the direction of its axis remaining fixed. The rotation in the opposite direction will be

$$b\ d = a\ b \sin \beta.$$

Therefore the effective rotation is

$$a\ b\ (\sin \beta - \sin \alpha).$$

In Fig. 136 A and B are the two arms of the planimeter, c the fixed point and d the tracing point. Let $a\ d$ make with the line $a\ b\ m$ the angle β. Draw an arc $d\ f$ and let the angle $f\ a\ m$ equal α, then keeping $a\ f$ parallel to itself, move it into the position $b\ g$. Draw now the arc $g\ e$ and finally move $b\ g$ back to its initial position $a\ d$.

We have $a\,b = d\,e = f\,g$ and the area $d\,e\,f\,g$ is equal to $h \cdot a\,b$, where h is the distance between the two parallel lines $d\,e$ and $f\,g$.

Now $h = a\,d\,(\sin\beta - \sin\alpha)$, therefore the area $d\,e\,f\,g = a\,d \times a\,b\,(\sin\beta - \sin\alpha)$.

If now we make the pointer follow the boundary $d\,e\,g\,f\,d$ we see that during its passage from e to g the

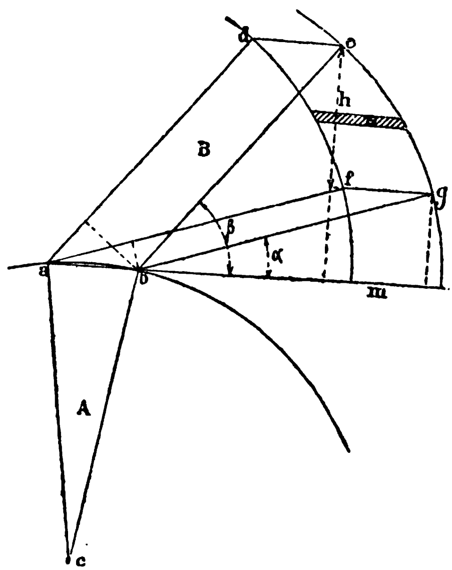


FIG. 136.

rotation of the wheel is equal and opposite to its rotation when moving from f to d. From d to e the rotation of the wheel is proportional to $a\,b\,\sin\beta$, and from g to f it is proportional to $a\,b\,\sin\alpha$; hence the total rotation is proportional to $a\,b\,(\sin\beta - \sin\alpha)$ and it is therefore proportional to the area $d\,e\,f\,g$.

Suppose now that we have to measure the area $a\,b\,m\,n$ (Fig. 137). We can divide it up into a series of strips whose boundaries are circular arcs traced by

the pointer of the arm *B*. By what we have just shown if we make the pointer move round *a b c d a*, the rolling wheel will register an amount proportional to the area of this element. In the same way we shall get the area of the second element by moving the pointer over *a d e f g h a* and so on. But we have seen that the path *b c e* can be replaced by *b e*; hence, in conclusion, if we make the pointer travel

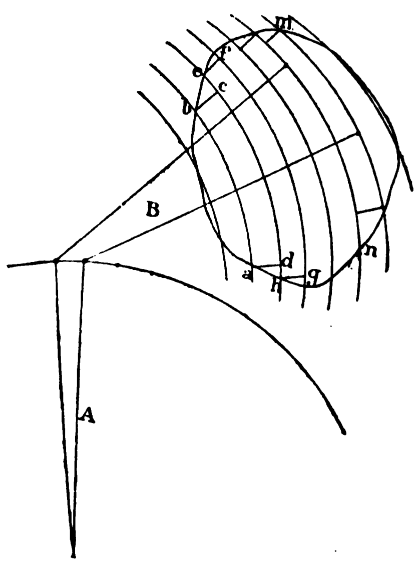


FIG. 137.

round the contour of the figure *a b e m n h a* the reading of the rolling wheel will always be exactly proportional to the area of the figure.

Planimeters can be marked so that they give readings in various scales.

If the area is too large for the instrument we subdivide it into several portions, find the area of these portions separately, and then their sum gives us the required area.

In special planimeters for indicator diagrams the arm B is made proportional to the breadth of the diagram, so that we can obtain the mean height directly.

Mean Pressure.

Whatever the method employed to determine the mean height of the diagram (y_m), if e be the scale of the diagram, that is the deflection of the spring for 1 pound pressure per square inch of piston face, the mean pressure in lbs. will be given by

$$P_m = \frac{y_m}{e}$$

Revolution Counter (Fig. 138).

For slow speed machines we can count the number of revolutions per minute directly by means of a

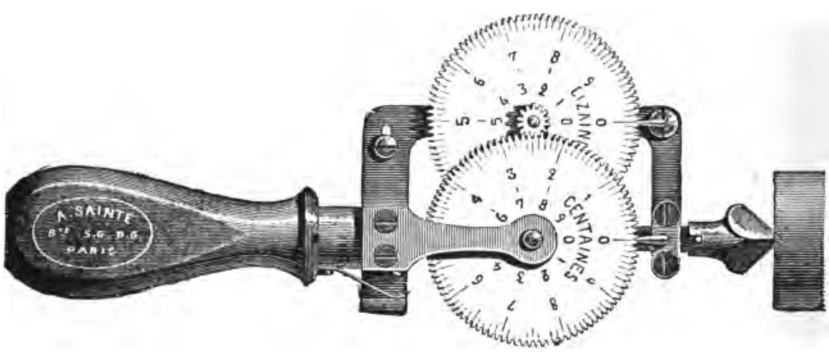


FIG. 138.

seconds watch. In general, however, the use of a revolution counter is more convenient.

If counting a few hundred turns is sufficiently accurate for the test, then we can use a simple counter which is pressed by hand against the end of the axis

of the rotating shaft, and we count the minutes by means of a watch. The counter of A. Sainte (Fig. 138) is very convenient for this purpose. The triangular point is pressed firmly against the end of the axis of the shaft whose revolutions have to be counted. An endless screw causes the wheel which registers the tens to rotate, and a little pinion fixed on the axis of this wheel makes the wheel which counts the hundreds turn round.

For a new reading we turn the zeros on each wheel opposite their indexes. To do this it is sufficient to press the bridge supporting the first wheel with a finger, which puts the pinion out of gear and so both wheels can easily be set. When the pressure is removed it is brought back into gear by means of a little spring.

For prolonged tests we use counters permanently fixed to the shaft.

Calculation of the Indicated Power.

Let p = the mean pressure of the steam in pounds per square inch on the piston face.

A = the area of the piston face in square inches.

D = the diameter of the piston face in inches.

N = the number of revolutions per minute of the fly wheel.

L = the stroke of the piston in feet.

V = the mean velocity of the piston in feet per minute.

$= 2\,N\,L.$

The indicated horse power will be given by the formula—

$$\text{I.H.P.} = \frac{2pLAN}{33000}$$
$$= pN\left(\frac{2LA}{33000}\right)$$
$$= pNV.$$

where $V = \dfrac{2\,LA}{33000} = 0{\cdot}0000606\ LA$ and is constant for a given engine. We must be careful to take for A the mean of the areas of the two sides of the piston, deducting the space occupied by the piston rod.

Example.

Let $D = 14$ inches; $A = 154$ square inches.
$L = 4{\cdot}5$ feet; $N = 200$.
$p = 30{\cdot}5$ lbs. (found from the diagram).

Then
$$\text{I.H.P.} = \frac{2\,pLAN}{33000}$$
$$= \frac{2 \times 30{\cdot}5 \times 4{\cdot}5 \times 154 \times 200}{33000}$$
$$= 256$$

Example of Compound Machine (Figs. 110, 111).

Replacing p by $\dfrac{y_m}{e}$ the formula becomes

$$\text{I.H.P.} = \frac{2LA}{33000e} \times y_m \times N$$

From the dimensions of the cylinders given in Chapter V. we find—

		Front.	Back.
A = useful area in square inches	H.P. cylinder	95	98
	L.P. cylinder	277	280

e = the scale of the springs used:

H.P. cylinder		$\frac{1}{38{\cdot}8}$	$\frac{1}{38{\cdot}2}$
L.P. cylinder	pressure	$\frac{1}{15{\cdot}9}$	$\frac{1}{15{\cdot}7}$
	vacuum	$\frac{1}{16{\cdot}3}$	$\frac{1}{16{\cdot}1}$

$\dfrac{LA}{33000e}=$

H.P. cylinder		1·74	1·81
L.P. cylinder	pressure	2·12	2·15
	vacuum	2·17	2·19

Multiplying the last numbers by y_m N where y_m is the mean ordinate found from the diagrams drawn in Figs. 110 and 111, and N is the number of revolutions given by the counters, we obtain the following numbers for the indicated horse power H'.

The brake horse power H being determined also by one of the methods we shall describe shortly, we can find the efficiency $\eta=\frac{H}{H'}$ of the engine.

On no load $H'=6$ and $\eta=0$.

$$\text{First trial}\ \left.\begin{matrix}H'=56\\ H=48\end{matrix}\right\}\ \left.\begin{matrix}\text{frictional losses}\\ 8\end{matrix}\right\}\ \eta=0{\cdot}857=85{\cdot}7\%$$

$$\text{Second trial}\ \left.\begin{matrix}H'=69\\ H=60\end{matrix}\right\}\ \left.\begin{matrix}\text{frictional losses}\\ 9\end{matrix}\right\}\ \eta=87\%$$

$$\text{Third trial}\ \left.\begin{matrix}H'=80\\ H=69\end{matrix}\right\}\ \left.\begin{matrix}\text{frictional losses}\\ 11\end{matrix}\right\}\ \eta=86{\cdot}3\%$$

For large machines in good working order we can assume that η lies between 80 and 90 per cent. For locomotives we assume that the useful power at the rim of the wheel equals 80 per cent. of the *I.H.P.* A knowledge of the approximate value of η enables us

to find roughly the brake horse power in those cases where a brake test is impossible.

If the power expended on the piston be calculated (H_1) instead of being deduced from the diagram (H') the ratio $\frac{H}{H_1}$ is always less than $\frac{H}{H'}$, and this ratio varies with the power of the engines as follows:

Value of η for engines with ordinary expansion.

Power		4-8	8-15	15-25	25-40	40-60	60-80	80-120
$\eta=\frac{H}{H'}$		"	"	0·70	0·75	0·78	0.80	0·82
$\eta=\frac{H}{H_1}$	condensing	0·40	0·45	0·52	0·58	0·64	0·70	0·75
	non-condensing	0·45	0·52	0·60	0·65	0·70	0·75	0·80

Calculation of the Diameter of the Piston.

The values of η given at the end of the preceding paragraph enable us to calculate the diameter of the piston of an engine working under given conditions. We can determine p from the theoretical diagram. We have, if H be the brake and H' be the indicated horse power,

$$H=\eta H'$$

$$=\eta\frac{2pL\times 0\cdot7854D^2N}{33000}.$$

$$D=\sqrt{\frac{21000\,H}{PLN\eta}}=145\sqrt{\frac{H}{pLN\eta}}$$

$$=205\sqrt{\frac{H}{pv\eta}}$$

where v is the mean velocity of the piston in feet per minute.

Example.

Suppose that we have to design a condensing engine which will satisfy the following conditions:

$H = 100$ horse power.

$p = 30{\cdot}1$ lbs. per square inch.

$v = 360$ feet per minute.

$\eta = 0{\cdot}80$ from the table given above.

$$\text{Hence } D = 205\sqrt{\frac{100}{30{\cdot}1 \times 360 \times 0{\cdot}80}}$$

$$= 22 \text{ inches.}$$

Work done against Friction.

This work is the difference $H'-H$ between the brake horse power and the indicated horse power. The application of a brake to commercial engines is not always possible, and hence it is useful to be able to calculate it approximately. This can be done by the following method. Take the indicator diagram on no load, and calculate the work required to be given to the fly-wheel. When taking the diagram the engine must be disconnected from the external shafting. If there is still some gearing which the engine drives, the work done on it must be roughly calculated.

The work done against friction measured in this manner must be a minimum value, for we have neglected the friction of the packing rings round the piston.

Diagrams on no Load (Fig. 139).

Although the diagram gives us the work done by the steam on the piston at no load, yet it is necessary

to make sure that there is no leakage of steam round the piston. In order to do this we move the crank pin into its dead points before we admit the steam into the cylinder. Then, opening the cocks on both sides of the piston, we can make sure that there is no leakage of steam in the cylinder or in the pipes connected with it.

In condensing engines we can find out if there is leakage by putting on the exhaust pipe a metallic rod,

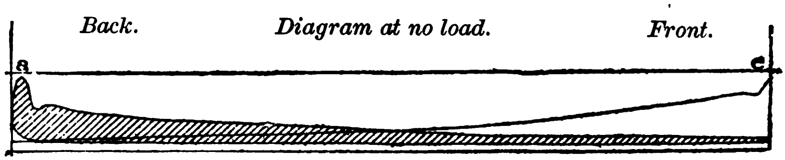


FIG. 139.

the other end of which is held between the teeth. On stopping up our ears we can hear when there is no leakage, the steam going in jerks into the condenser; in other cases the noise is more prolonged, and the difference can easily be distinguished after a little practice.

It is necessary also to keep the machine running at its normal velocity, and as far as possible to arrange so that the frictional pressures on the moving parts are the same as when the engine is loaded, in order that the work calculated from the diagram may include all frictional losses. This condition cannot always be arranged in engines which have no variable expansion gear.

Inertia of the Fly-wheel.

When the machine is running at its normal speed of N revolutions per minute the steam cock is closed and the time (t) in seconds taken by the fly-wheel to come to rest is observed by means of a stop-watch.

The angular velocity (ω) of the fly-wheel initially equals $\frac{2\pi N}{60}$ and can be found.

The energy E in foot-pounds stored up in it is given by the formula:

$$E = \frac{MK^2}{2g}\omega^2$$

where M is the mass of the fly-wheel in pounds, K its radius of gyration, and g is the acceleration produced by gravity, and it equals 32·2 in London. The average power exerted by the frictional forces in stopping the fly-wheel will then be

$$\frac{E}{550\,t}$$

In the above calculation we have neglected the inertia of the pulleys, gearing, etc. This could be taken into account by adding on to $M\,K^2$ in the above formula, $M_1\,K_1^2 + M_2\,K_2^2 + \ldots$ where M_1, M_2 and K_1, $K_2 \ldots$ are the masses and the radii of gyration of the other rotating bodies.

The kinetic energy of rotating shafts is always very small, and can generally be neglected.

The moment of inertia of the rim of a fly-wheel whose mass is M, and inner and outer radii R and r respectively, is:

$$M\frac{R^2 + r^2}{2}$$

The moment of inertia of the spokes of length l is

$$M'\frac{l^2}{3}$$

where M' is their mass.

Example.

Suppose that the fly-wheel is making 60 revolutions per minute, and that it stops in 660 seconds when the steam is cut off.

Let $M = 20{,}000$ lbs.; $R = 10$ feet; $r = 8$ feet.

$M' = 6{,}000$ lbs.; $l = 8$ feet.

We shall have

$$\omega^2 = \left(\frac{2\ \pi\ 60}{60}\right)^2 = 39{\cdot}48$$

$$\therefore\ E = \frac{1{,}768{,}000}{64{\cdot}4} \times 39{\cdot}48 \text{ foot-pounds.}$$

$$= 1{,}084{,}000 \text{ foot-pounds.}$$

$\therefore$ Mean frictional horse power

$$= \frac{1{,}084{,}000}{550 \times 660}$$

$$= 2{\cdot}986.$$

CHAPTER VIII.

BRAKE HORSE POWER.

§ 1. ORDINARY BRAKES.

THE brake illustrated in Fig. 140 was constructed in 1821 by Prony. The power generated by the motor is expended in overcoming the friction of a collar placed round a pulley, and the amount of the work done can be calculated from the force required to keep this collar in its place.

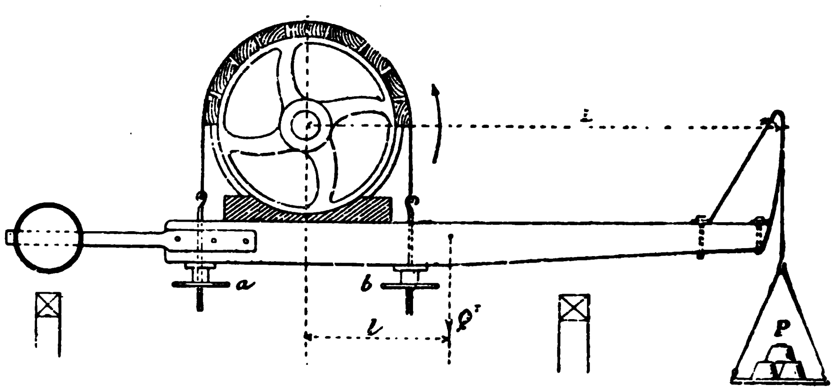


FIG. 140.

This brake can be applied to any motor whatever which produces a motion of rotation.

The collar or chain of wooden blocks can be put directly on any convenient turning part of the shaft when we are measuring small horse powers. The

brake is usually applied to the pulley of the motor. It is, however, convenient to have a special pulley which will fit any size of shaft.

The large lever is placed preferably underneath so as to increase the stability, and a counterweight is added to it so that the centre of gravity of the whole may come directly underneath the axis. The oscillations of the lever are limited to an inch or so by means of stops.

If we place at the extremity of the lever (Fig. 140) an arc of a circle whose centre is in the axis of the pulley, the weight *P* will always act at the same distance *L* from the axis and the method will be more exact.

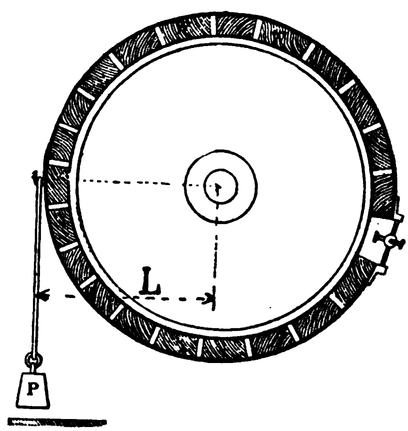


Fig. 141.

The simple brake shown in Fig. 145 can be made anywhere. The two blocks of wood applied to the pulley are kept in position by two bolts which are joined above and below by iron bands.

In Fig. 141 the brake band consists of wooden blocks, and it can be tightened by means of screws. The weight *P* is hung from a hook fixed on one of the blocks. The friction is in this case distributed over a very large surface.

For small horse powers the wooden blocks can be replaced by a copper band, which should be worked without the application of oil.

Vertical Shaft (Fig. 142).

The horizontal lever has a rope attached to its extremity which passes over a vertical pulley and carries the pan for the weights. Sometimes, however, as in Fig. 142, the rope is attached to an oscillating triangle, which carries on one side the scale pan and on the

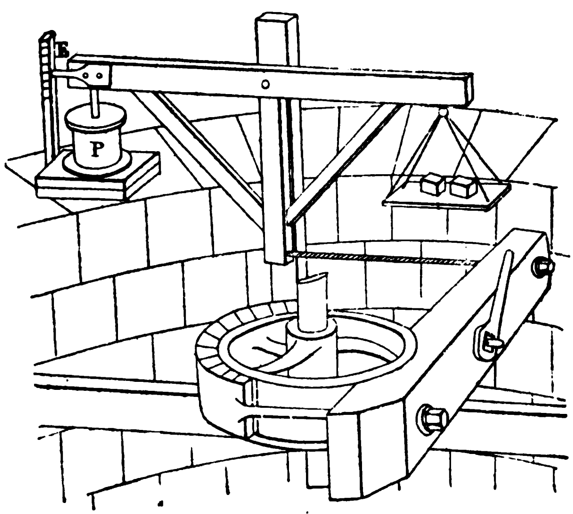


FIG. 142.

other a piston passing into a dash-pot *P* to prevent oscillations, and an index which moves along a scale *E*.

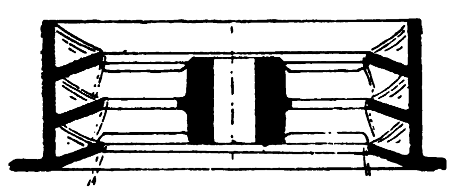


FIG. 143.

The pulley can be covered underneath so that it forms a reservoir in which water can be kept circulating for cooling purposes.

In Fig. 143 another arrangement is shown. The pulley is provided with ribs in its interior, which are slightly inclined to the horizontal. When water is poured from above the centrifugal force makes it flow outwards, and it is kept pressed against the rim of the wheel, where it descends step by step, keeping the surface cool. When the velocity is high the ribs can be horizontal, as the centrifugal force is sufficient to keep the water pressed against the rim.

Procedure during a Test.

The brake being mounted and the weight P being in position, we start the machine. The brake is gradually tightened by means of the nut a, which is on the side of least tension (Fig. 140). We then increase or diminish the weight P until the brake is in equilibrium when the motor is running with its normal speed.

When the brake is working we can cool the surfaces on which the frictional forces are acting by means of a trickle of water containing about ten per cent. of dissolved soft soap. To get a good solution soft water must be used. We employ also grease, lard, etc., or better, the more fluid oils, but they ought not to be used twice without being refiltered. The lubricant must not be altered during the trial, otherwise there is a risk of the friction becoming abnormal during the change, the brake seizing and the equilibrium being destroyed.

The speed of the motor and the friction both vary during a trial. We restore a balance at the normal speed by screwing or unscrewing the nut a (Fig. 140)

and by altering the weight P. We can obviously also obtain the work done at various speeds, and so determine at what speed certain motors—like turbines, for example—do their maximum work.

If it is not necessary that the motor run at a fixed speed, and we obtain a balance by simply screwing and unscrewing the nut without altering the weight P, then the revolution counter will register the total number of turns made by the pulley during the test. In order to obtain a constant load it is necessary to maintain the temperature of the brake constant. We can effect this by regulating the flow of the soapy water. If this means is not sufficient, we can inject cold water into the interior of the pulley. This will in general reduce the flow of the lubricant.

Volume of Water Required.

The temperature caused by the frictional forces will be higher the greater the pressure per square inch of the rubbing surfaces. Each wooden block, therefore, ought to press on as large a surface of the pulley as possible.

A new brake always heats up more than an old one, because the surfaces of the blocks are not in such close contact with the pulley. The real surface of contact is less, and therefore the pressure per square inch is greater.

The mechanical equivalent of heat being 778, a horse power will develop in an hour

$$\frac{550}{778} \times 3600 = 2545 \text{ B. T. U.}$$

Suppose that $t°$F. is the initial temperature of the water and T is the temperature of the brake. Then if Q be the mass of water in lbs. required per horse power per hour—

$$Q = \frac{2545}{T - t}$$

Example.

Let $t = 45°$ F. and $T = 95°$ F.

then $T - t = 50$.

Therefore Q is 51 lbs. or 5·1 gallons of water per horse power per hour. The least possible quantity of water required will be that corresponding to the total heat of evaporation of water at 212° F., i.e.

$$966 + 180 = 1146.$$

$$\text{Hence } Q = \frac{2545}{1146}$$

$$= 2{\cdot}2 \text{ lbs. nearly.}$$

In practice it is necessary to use twice or three times the quantity of water calculated from the above formula, as all the water does not attain to the temperature of the brake.

Calculation of the Brake Horse Power.

The weight P, including the weight of the scale pan, acts as if it were suspended by a weightless cord hung round a cylinder of radius L (Fig. 140). The balance being established and the speed uniform, the brake H. P. is given by the formula

$$\text{B. H. P.} = \frac{T\,\omega}{550}$$

where T = the torque = $P\,L$ foot-pounds.

ω = the angular velocity.

$$= \frac{2\ \pi\ N}{60}$$

where N = the number of revolutions per minute.

Therefore the

$$\text{B. H. P.} = \frac{2\pi PLN}{33000}$$

$$\text{and } H = 0{\cdot}0001904\ PLN \quad . \ . \ (1)$$

if H stand for the B. H. P.

$$\text{Hence also } P = \frac{5250\ H}{L\ N} \quad . \ . \ . \ . \ . \ . \ . \ (2)$$

Knowing the speed of the engine (N), the length of the lever of the brake (L) and the probable brake H. P. this formula will enable us to find P.

If when the pulley is at rest the centre of gravity of the beam is not directly under the axis of the pulley (Fig. 140), a correction must be applied to the formula. Suppose, for example, that the weight of the beam Q' acts at a distance l from the vertical line through the axis of the pulley, then the weight Q, which, added to P, would produce the same torque, is given by—

$$Q = Q' \frac{l}{L}$$

It is also easy to find Q directly. If the beam be supported by a round or angular bar placed underneath it at the point where the vertical line through the axis of the pulley meets the surface of the beam, and if the end of it be supported by the pan of a spring balance, then the reading on the balance will give Q.

This weight Q, which takes into account the weight of the beam itself, has to be added or subtracted from

P, according as the rotation of the pulley tends to raise or lower the beam. The formula (1) now becomes

$$H = 0{\cdot}0001904\,(P \pm Q)\,L\,N \;.\;.\;.\;.\;.\;.\;.\; (3)$$

Calculation of the Dimensions of the Brake.

Pulley.—The dimensions of this vary with the heating, and this varies with the amount of the lubricant and with the quantity of water injected into the interior of the pulley. It also varies according as the brake embraces the whole circumference of the pulley or a part of it only.

Let D = the diameter of the pulley in feet.
l = the breadth of the brake in feet.
H = the horse power measured.
K = the number of foot-pounds per second absorbed by the friction per square foot of surface.

If the brake completely embraces the pulley, and if it is kept cool by the injection of water into the interior of the pulley, then we can use for large horse powers the following approximate formulae :—

$$D\,l = \frac{H}{50} \text{ or } D\,l = \frac{H}{40}$$

To find the number of foot-pounds per second absorbed by a square foot of surface, we have—

$$K \times \pi\,D\,l = 550\,H.$$

Putting $D\,l = \dfrac{H}{50}$

we find that

$$K = 9000 \text{ approximately.}$$

We shall see later that the Thiabaud brake (Figs. 148,

149) absorbs nearly 15,000 foot-pounds of work per second per square foot of brake surface.

When only a limited amount of water can be applied it is better to use the formulae—

$$D\,l = \frac{H}{30} \text{ or } D\,l = \frac{H}{20}$$

The first equation gives $K = 5400$ foot-pounds. When the brake envelops the whole surface of the pulley and is cooled only by the lubricant it is necessary to make the pulley larger. In this case

$$D\,l = \frac{H}{10} \text{ or } D\,l = \frac{H}{8}$$

Finally, if the brake envelopes only a portion of the surface of the pulley, it is necessary to still further increase its dimensions.

For a given horse power the dimensions of the pulley are independent of the speed of the motor, as the frictional work per second on its circumference is the same at all speeds.

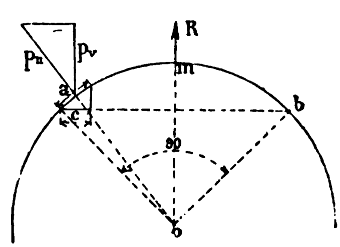


FIG. 144.

Bolts.—In order to find the dimensions of the bolts we will consider the equilibrium of the upper block (Fig. 145). Screwing up the nuts increases the small elementary normal pressures p_n in pounds weight on the pulley rim, and these produce frictional forces $f\,p_n$.

$$\text{Let } F = f \Sigma p_n$$

be the sum of the frictional forces on the block. Since the brake is in equilibrium, we have by taking moments for the two blocks—

$$2 F r = P L$$

Also equating the work per second

$$2 F v = 550 H,$$

where v is the velocity of the rim of the pulley in feet per second.

$$\text{Hence } F = 275 \frac{H}{v}$$

$$= 275 \frac{H}{\frac{2 \pi N r}{60}}$$

$$= 2600 \frac{H}{r N} \text{ nearly} \quad . \quad . \quad (4)$$

Also taking $f = 0{\cdot}2$ we find that

$$\Sigma p_n = 13000 \frac{H}{r N} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

We still have to determine the vertical resultant R.

Considering one of the elementary normal pressures p_n acting on an element 'a' of the arc $a\, m\, b$ (Fig. 144), p_v is the vertical component, and 'c' is the projection of 'a' upon the chord $c\, b$. From the two right-angled triangles thus formed we get

$$\frac{p_n}{a} = \frac{p_v}{c}$$

$$\text{Therefore } p_v = \frac{c}{a} p_n$$

$$\text{Hence} \qquad R = \Sigma \frac{c}{a} p_n$$

$$= \Sigma\, p_n \times \frac{\text{chord } c\, b}{\text{arc } c\, b}$$

Suppose the arc subtends an angle of 90 degrees at the centre of the pulley, then

$$\text{the chord } a\, m\, b = \frac{\pi}{4} D$$
$$= 0{\cdot}7854\, D$$
$$\text{chord } c\, b = \text{diagonal of square}$$
$$= 0{\cdot}7071\, D$$

$\Sigma\, p_n$ is therefore given by the formula

$$R = \frac{0{\cdot}7071}{0{\cdot}7854} \times 13000 \frac{H}{r \cdot N}$$
$$= 11700 \frac{H}{r \cdot N} \quad \ldots\ldots\ldots \quad (6)$$

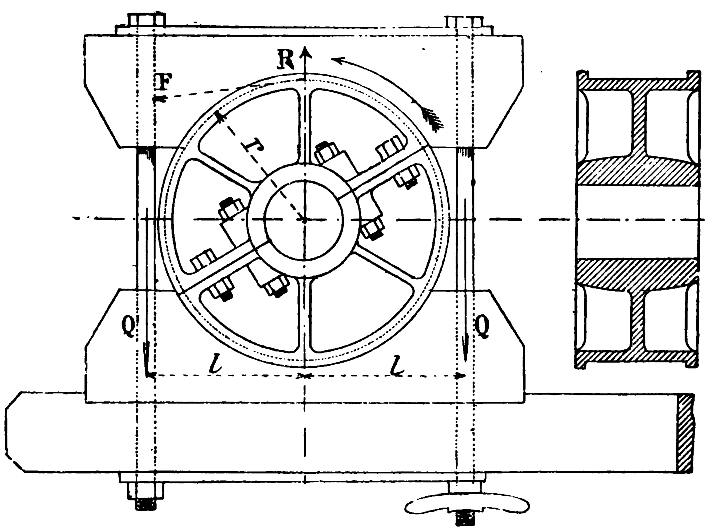


FIG. 145.

Now writing the conditions for equilibrium,

$$Q + Q' = R$$

And taking moments $l\, Q - l\, Q' = F\, r$

$$\therefore \quad Q - Q' = F\frac{r}{l}$$

Hence $Q = \frac{1}{2}\left(R + F\frac{r}{l}\right)$ . . . . (7)

and $Q' = \frac{1}{2}\left(R - F\frac{r}{l}\right)$ . . . . (8)

Substituting the values of R and F from (4) and (6) in (7) and (8) and simplifying, we get approximately—

$$Q = 1250\,\frac{H}{N}\left(\frac{5}{r} + \frac{1}{l}\right)$$

$$Q' = 1250\,\frac{H}{N}\left(\frac{5}{r} - \frac{1}{l}\right)$$

Example.—Suppose that the ratio of H to N is 2, that the radius of the pulley is 2 feet, and that l equals 2 feet 6 inches, we shall have

$$Q = 1250 \times 2\left(\frac{5}{2} + \frac{1}{2 \cdot 5}\right)$$

$$= 7{,}200 \text{ nearly.}$$

Supposing then that the working load on the tight bolt is to be 10,000 lbs. per square inch, we see that a bolt whose diameter at the bottom of the threads is one inch will be ample. The two bolts are, of course, taken of the same size.

Beam.

The weight P required in the given test which can be calculated *a priori*, enables us to calculate the section $a \times b$ of the beam at the bolt Q.

The bending moment $= P\,(L-l)$

$$= R\,\frac{ab^2}{6}$$

R, the stiffness, being 120,000 pounds per square foot.

Example.

Suppose that $L = 10$ feet, $l = 2$ feet, $H = 100$, and $N = 50$

Then from (2)

$$P = \frac{5250\ H}{L\ N} = 1000 \text{ pounds weight nearly.}$$

If the breadth (a) of the beam is 8 inches, then its thickness b can be found by means of the above formulae as follows:— $b^2 = \frac{6 \times 10008}{120,000 \times \frac{2}{3}}$

$$\therefore \frac{3}{5}.$$

$$\therefore b = 0{\cdot}775 \text{ of a foot.}$$

$$\doteq \text{nine inches nearly.}$$

Detachable Brake.

This brake was constructed for the rapid testing of machines at the Dusseldorf Exhibition. The tightening of the brake can be done by means of an endless screw which acts on the slacker of the two bolts. This bolt is connected to the flexible band by means of a system of jointed levers which allows the brake to be rapidly put into or taken out of action. The extremity of the long lever acts on the platform of a weighing machine situated in the horizontal plane passing through the axis.

The reading of the weighing machine gives the total pressure, including the weight Q, due to the beam not being balanced initially. It is therefore necessary to find Q first of all, and then subtract it from the reading of the machine P, so as to find the P–Q of formula (3). The slacker of the two bolts works on

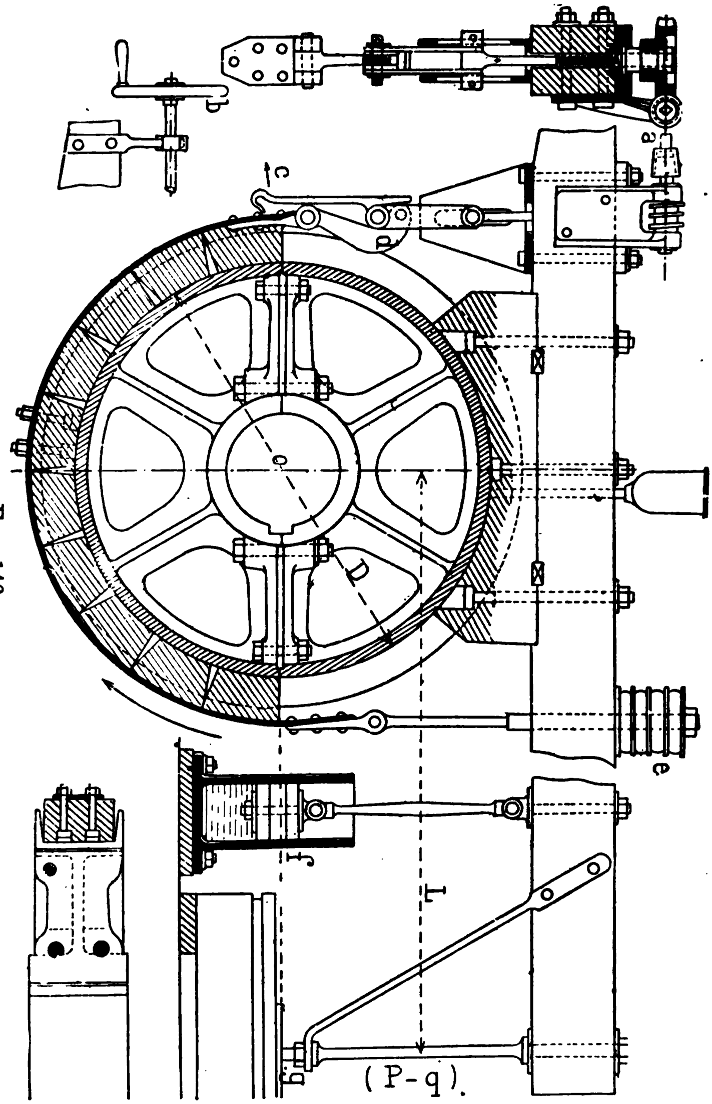


FIG. 146.

elastic washers, which ensures smooth running during the trial.

When the oscillations produced are too large, the apparatus can be completed by an hydraulic dash pot

consisting of a piston placed in water. So long as the movements of the piston are gentle the water has time to pass from one face of the piston to the other; for jerky motions, however, the water offers a great resistance to the motion of the piston.

A drawback to the use of this brake is the inequality of the heating of the frictional surfaces. The upper block has a smaller surface in contact with the wheel than the lower band, and hence the pressure per square inch and the heating are greater than for the band.

Société Centrale Brake.

The faces of this pulley (Fig. 147) are enclosed by iron plates.

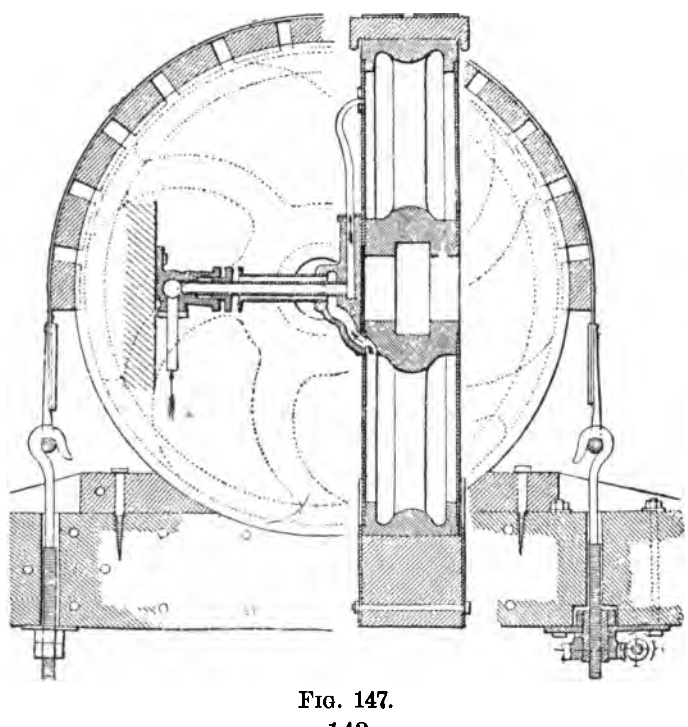


FIG. 147.

The water coming from a high level reservoir flows by means of a central tube near the rim into the interior of the pulley, which it cools, and then flows out as shown in the figure. The brake is tightened by means of a nut operated by a wheel, and it works on the bolt on which there is the greater pull.

It is necessary to put this brake on the extremity of the shaft of the motor, and this is not always possible.

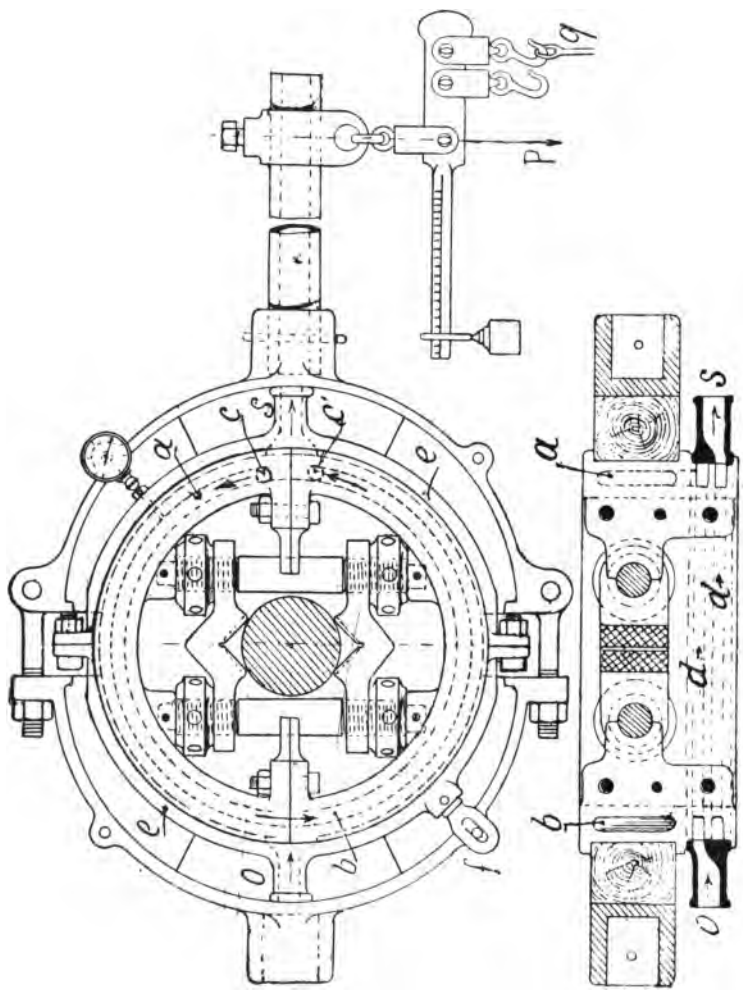
The Thiabaud Brake (Figs. 148, 149).

In this brake, which is employed by the Italian Government, the water circulates in the pulley itself.

The pulley, which is hollow, is divided into two parts, and is centred on the shaft (Fig. 148) by means of two V-shaped pieces of metal, which clutch the shaft and can be tightened by four nuts. The enveloping brake is formed of two semicircles of iron fitted with wooden blocks. The lever consists of an iron rod, which can be fixed on the side which is the more convenient. The pull P is measured by a Roman steelyard, the end of which is fastened to the ground by a rod or cord q.

The pulley has an interior channel a b, which is in communication during half a revolution with one of its junctions b, the other junction a being closed. One side communicates at c with an exterior opening d drilled in the pulley; the other side communicates with c' by means of the opening d'. The openings are covered by a collar e made in two pieces, which are fixed firmly to the brake by the gudgeon f, and it carries two tubes, o and s. If then cold water flows

through the tube o, it will always penetrate by the opening d and the orifice c, notwithstanding the rota-



Figs. 148, 149.

tion of the pulley, and will come out by the opening d' and the orifice c' after having been carried round once.

A thermometer enables us to read the temperature of the water coming out of the pulley. The friction

of the collar e e is added to that of the brake, but the pull P takes this into account.

The following results have been obtained with this brake:

Diameter of pulley A in inches	12	15	18	22
Brake Horse Power	15	20	25	50

With the largest pulley—taking its breadth to be one-sixth its diameter—we find that the foot-pounds of work per second dissipated by a square foot of the surface of the brake are about 15,000.

Carpenter's Hydraulic Brake (Figs. 150–152).

The novel brake represented in Fig. 150 was invented by Professor Carpenter (see *Engineering*,

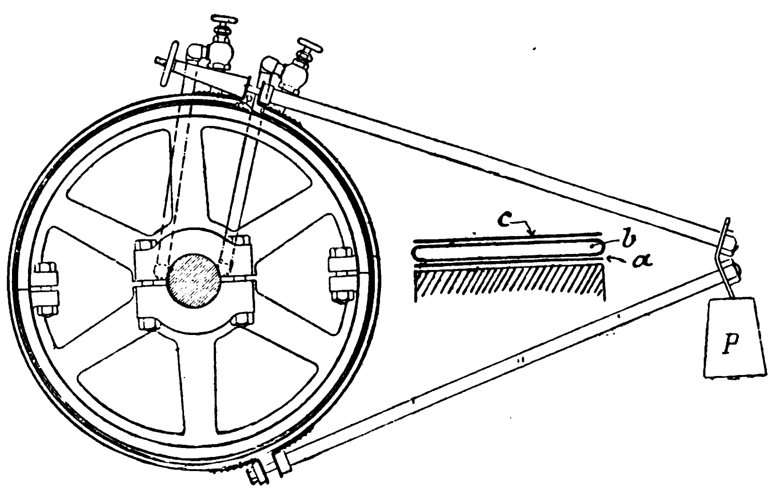


Fig 150.

January, 1894). The band of the brake which envelops the pulley is made up (1) of a simple flexible steel band a on which the frictional forces act; (2) of

a thin beaten copper tube *b* in which water is injected by a force pump; and (3) of an exterior band of steel *c* made in two parts, which are fastened to the bars forming the lever of the brake. The brake is con-

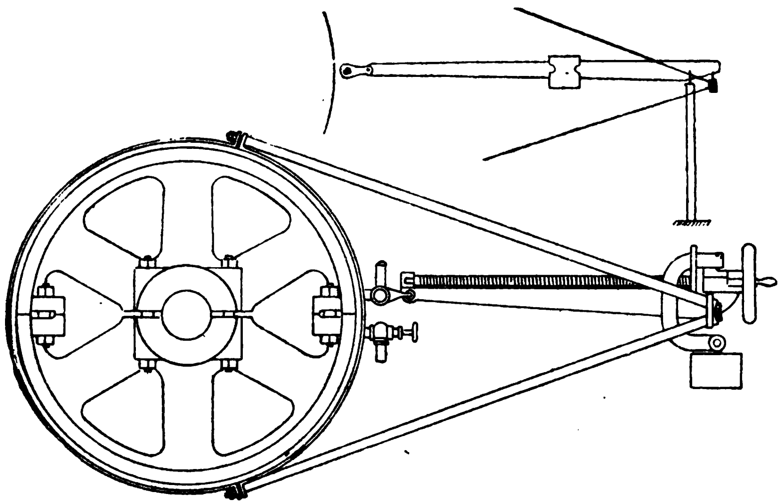


FIG. 151.

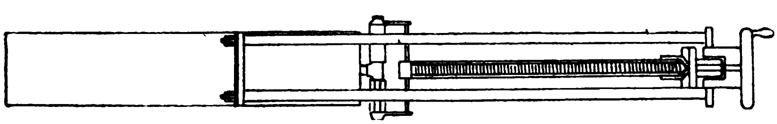


FIG. 152.

strained to oscillate within narrow limits. The high pressure water supply is led to the copper tube by means of flexible tubes, and it can be regulated by means of stopcocks. Since the diameter of the copper tube cannot vary, as it is pressed by the exterior steel sleeve, it follows that the effect of the hydraulic pressure is to press the interior face of the tube against the inner band, and so it increases the friction, and consequently the power absorbed. At the same time the circulating water carries away the

heat produced by the friction. It is in order to avoid the wearing away of the inner tube by friction that the flexible steel band *a* is interposed between it and the face of the pulley; being prevented from turning, it has to support all the frictional wear.

The lever of the brake can act upon the little arm of a steelyard (Fig. 152) and the cursor can be moved by means of a screw.

§ 2. AUTOMATIC BRAKES.

In these brakes once the adjustment is made the balance is maintained by means of the movements of the brake itself.

Brake with Spring Balance (Fig. 153).

We add a spring balance (Fig. 153) in opposition to the spring in the ordinary arrangement. The

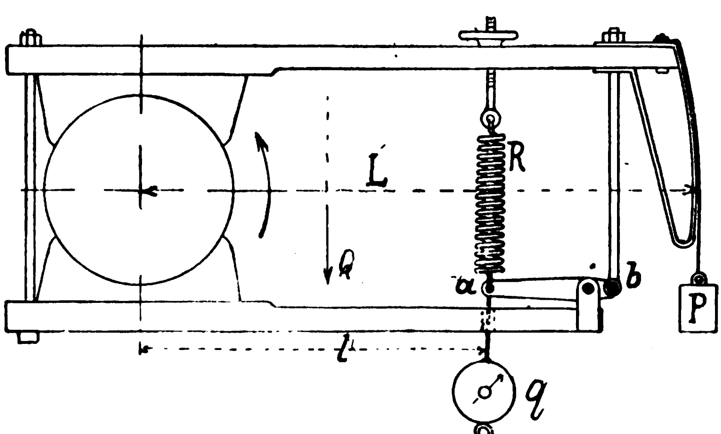


FIG. 153.

tension of the spring *R* adjusts itself corresponding with the equilibrium of the brake, and any excess is

measured by the spring balance q which is fixed to the ground. When the apparatus is in equilibrium the brake horse power can be calculated by adding to P the component due to q, Q being the weight required to be added to P owing to the moment of the weight of the brake itself about the axis of the pulley. We have

$$H = 0\cdot0001904\ (P + Q + q\frac{l}{L})\ L\ N$$

If the friction increase owing to want of lubricant the brake is dragged in the direction of the arrow, but then the tension q increases and the point a is lowered, thus loosening the brake. Conversely if the friction diminish the brake is tightened.

Creuzot's Arrangement (Fig. 154).

The clamping wooden blocks M (Fig. 154) each embrace a quarter of the circumference of the pulley. They are bolted to two iron bars which are connected at the back by two rods joined by a tension screw v which can regulate the distance apart of the blocks. In front the lower beam is connected to b in a similar manner, whilst the upper block supports the lever B at o. This lever B is fastened to a spring at its other extremity q and carries a pan for weights p.

Suppose, for example, that the point q is fixed, we see that the point o rises, the point b rises still further, and the brake tightens. On the other hand, if o falls, then b falls still further, and the brake is loosened.

The bar of the lower block which forms the large lever of the brake is attached to a scale pan P by means of a cord passing over a pulley. Pure water

is applied to the inside of the brake pulley and soapy water is applied outside.

The spring q exerts a vertical force represented by q, and hence the tightening force on the brake equals

$$p\frac{a}{b} - q.$$

where p is the weight on the pan p.

We see, if the rotation of the brake is in the sense

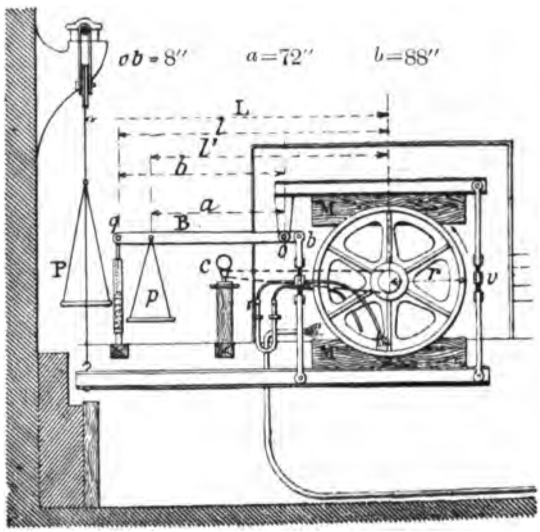


FIG. 154.

indicated by the arrowhead, that when P is too small the brake is turned round, B lowers, and thus q increases, and the brake being loosened, equilibrium is restored.

If on the other hand P is too great, it tends to raise the brake and B; q then diminishes, and the brake tightens.

In the trial runs of a Corliss Engine of 100 to 150 horse power, reported on by M. F. Delafond in

the *Annales des Mines* for 1884 the weight p was varied from 176 to 726 lbs., whilst the tension of the spring varied from 22 to 220 lbs. The spring is measured and the tension q is indicated by a pointer which moves along a scale. In Fig. 154 C is a tachometer which gives at each instant the speed of the pulley.

When we run a trial we first of all raise B; this lets the machine start; then we lower B and put weights on the pans P and p until the brake is in equilibrium at the required speed.

Calculation of the Work.

We first of all gauge the brake by putting weights on P until there is equilibrium. These weights have to be subtracted from the value of P observed during the trial in order to get the real value of P to substitute in the formula. Taking moments about the axis of the pulley we get

$$Fr = PL + ql - pl'$$

and for the work,

$$Fv = F\frac{2\pi rN}{60}$$

$$= 550\ H$$

$$\therefore H = 0{\cdot}0001904\,(PL + ql - pl')\ N$$

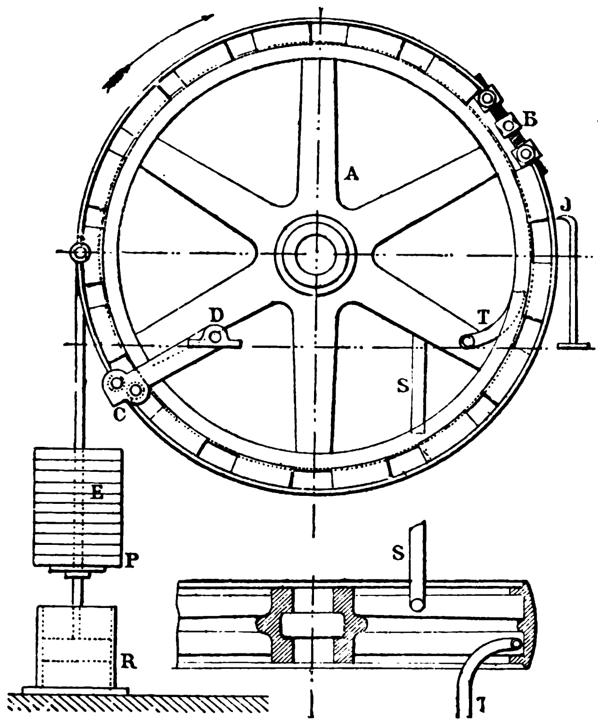
Amos or Appold Brake (Figs. 155–157).

This brake is used by the Royal Agricultural Society. The two halves of the brake are joined in one place by a tension screw B (Fig. 155), which adjusts the brake load at the start, and in another place C by two bolts unequally distant from the rim (Fig. 158), and are joined by levers which have slots D through which

passes a fixed axle. The bolt connecting the weight E to the brake band must meet it at the same point as the horizontal line through the axis of the pulley. This position is indicated by an arrow head which corresponds to the fixed index J.

FIG. 155

FIG. 156



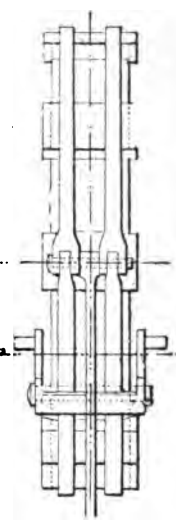


FIG. 157.

If there is too much friction the brake is dragged round in the direction of the arrowhead (Fig. 155), but the bolts C being raised up, slacken the brake and so restore equilibrium. Conversely if the friction be insufficient the bolts C are lowered and thus tighten the brake.

We can see what happens in another way by considering how the line joining the axes of the bolts C turns round the fixed axle at D. When C rises this line is less inclined to the tangent to the circumference of the pulley, and hence the brake is loosened, and when C falls this line approaches the normal and thus tightens the brake.

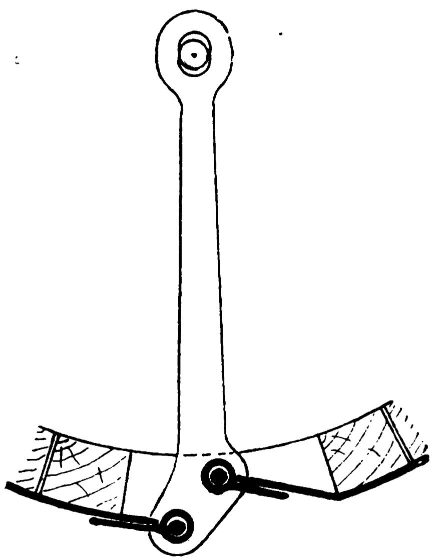


FIG. 158.

R (Fig. 157) is a water dash pot similar to those previously described.

When the brake is used to measure very small loads the oblique position of the tightening lever $C\,D$ is a drawback, as it produces certain frictional forces which cannot be calculated. It is better in this case, therefore, to place the lever vertically (Fig. 158).

The water cooling arrangement in the interior of the pulley is indicated in Figs. 155 and 157. The water enters by a tube S, and after getting heated it leaves by the bent tube T.

The small error due to the friction of D can be neglected in comparative tests of engines of about the same size. The friction due to this cause is about the one hundred and fiftieth part of the total friction.

The Balk Brake (Fig. 159).

The pulley, provided with its brake, is mounted on a wagon which allows it to be rapidly moved from one engine to another. The automatic tightening movement can be easily understood from the figure. The

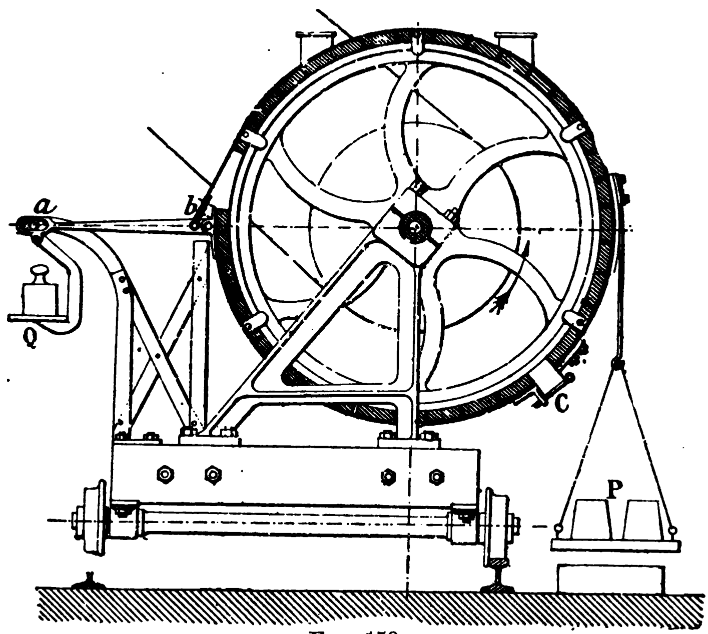


FIG. 159.

point *a* being fixed, if *b* is lowered by excessive friction the brake is automatically loosened and conversely.

We find the resultant of the forces at *b* by equilibrating them with a weight Q (Fig. 159). If then q be the distance of the weight Q from the vertical line through the centre of the pulley, and L be the distance of the line of action of P from the centre, then the

$$\text{Torque} = PL - Q\,q.$$

The Brauer Brake (Fig. 160).

The pulley is surrounded by an iron band, which is kept in position by guides *G* (Fig. 160).

This band is able to support twice the weight *P*

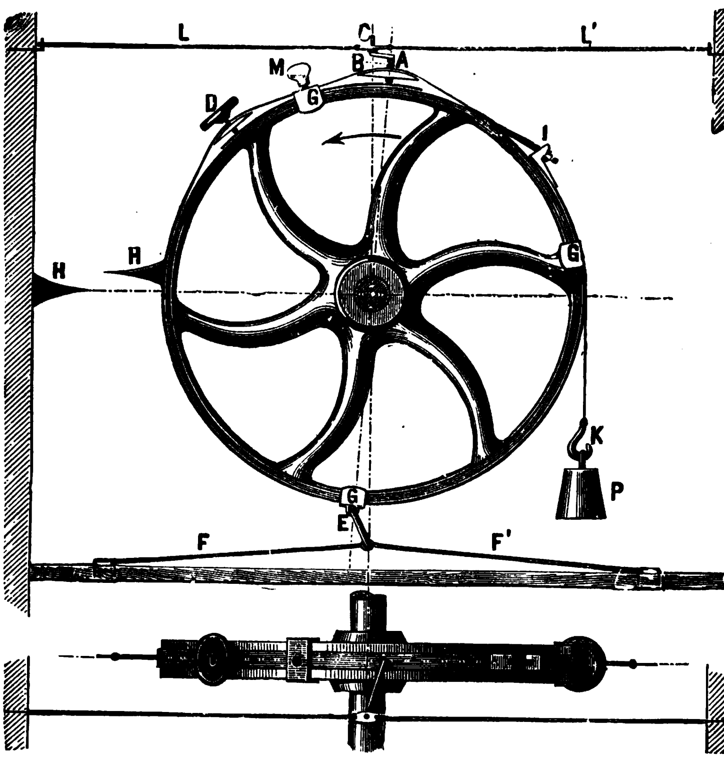


Fig. 160.

required to obtain a balance. The tightening screw *I* is only used when setting up the brake.

The lower guide *G* is connected by a link to two fixed cords *F* and *F'*; *D* is a tightening screw and *A* is another one with a handle *A C*. These screws, *A* and D, press upon a band of iron fixed on the top guide *G*, which also carries an oil cup. The tightness

of these screws are first regulated by hand. We then put the handle C through a hole in a plate of metal held fixed by the cords L and L'. The screw A must also lie on the vertical through the axis, and consequently the handle must be parallel to the axis of the pulley. When this has been done the cup is filled with oil, and we turn the pulley by hand to make sure that the brake is working properly, and we then start the engine.

So long as P is not hooked on to K, the cord F is stretched ; when we hang P on, the brake takes the position shown in the figure, and the cord F' is stretched. We now tighten the screw D until the friction causes equilibrium with the weight P, and the screw A is vertical when the indices H and H' are opposite one another. The weight P, which has been roughly calculated previously, is modified until we get a balance at the required speed. When this is the case the balance is maintained automatically.

If the friction is insufficient the brake tends to take the position shown in the figure, but the screw A now tightens, and equilibrium is established.

On the other hand, if the brake be dragged round the handle C passes to the right of A, and the tension is relieved, diminishing the friction and restoring the balance. The screw A oscillates from the right to the left of the vertical line through the axis of the pulley.

In consequence of the difficulties which arise in attaching the cords L and L' and the errors that result from their tensions, this arrangement is only suitable for small motors.

Another Arrangement.

For engines of from 15 to 20 horse power M. Brauer employs the arrangement shown in Fig. 161.

The upper band of the brake carries an oil cup *O*, and has at its extremity a bent lever *E* and a sleeve *G* which carries the weight *P*. The lower band is

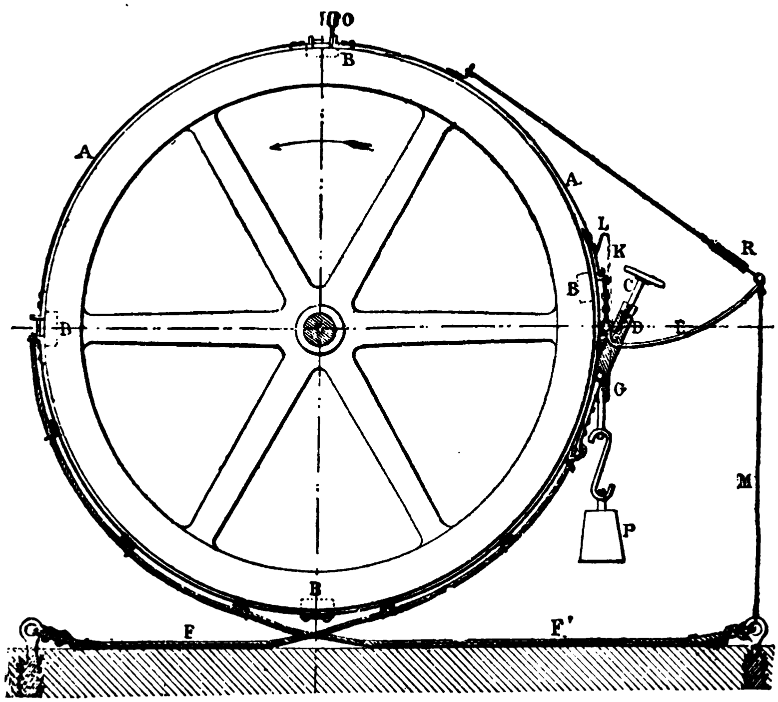


FIG. 161.

attached to the sleeve *D*, which forms the nut for the screw *C*. The brake can be tightened by means of the screw *C* which is fixed to the lever *E*, of which one part is attached to the band *A* by means of the spring *R*, and the other to a fixed point by means of the slack cord *M*. The safety ropes *F* and *F′* allow the brake a play of about four inches.

We can tell when the brake is in its mean position by means of the index finger *L*. The weight *P* is calculated roughly before we begin the test. The brake is free when the screw *C* is slack, and when the weight *P* is not hooked on. In this case the cord *M*

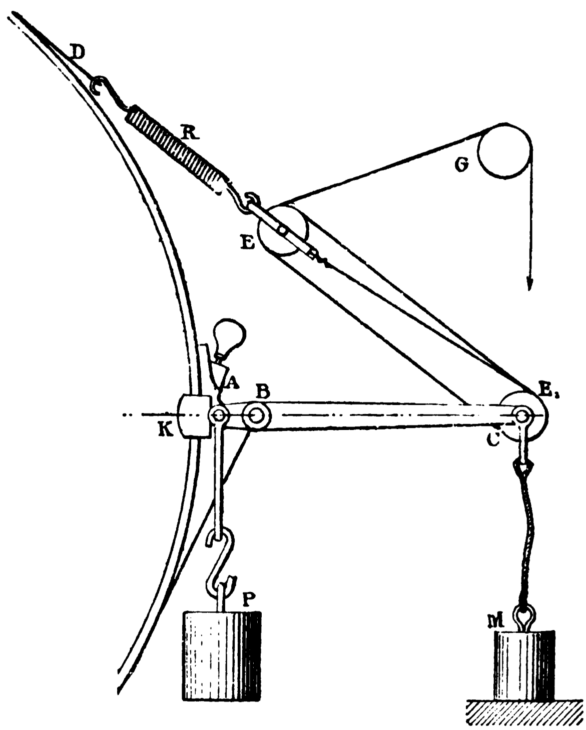


Fig. 162.

and the rope *F* are slightly stretched, and the machine runs freely. When *P* is hooked on the rope *F'* is stretched. We adjust the steam admission until it runs at its normal speed, at the same time turning the screw *C* until the friction is sufficient to support *P*. In this case the rope *F'* is slack.

Suppose now that owing to an increase in the value

of the friction the brake is dragged round with the pulley, the cord M becomes tight and slackens the brake. If, on the other hand, the weight P pulls it back, the cord M becomes slack and the spring R tightens the brake.

When equilibrium is established the tension of the cord M must be so small that it introduces no appreciable error in the calculation. In this, as in all other brakes, it is necessary to provide some cooling arrangement to prevent the temperature rising above 176° F. (80° C.), and to put the brake on as large a pulley as possible.

Upon grooved pulleys or flywheels we may replace the flat band of iron by iron wires, one in each groove, the diameter of the wires being proportional to the stress they will have to withstand. In this case it is necessary to have as many tightening levers as wires, and hence the arrangement is complicated. Owing to this complication it is sometimes better to use a flat band with grooved pulleys.

Another Arrangement.

For machines greater than 20 horse power the preceding arrangement can be modified as follows :—

The upper band is connected to the extremity of the lever $A\ C$ (Fig. 162), the lower band is jointed to the lever at B and the weight P is attached to A. Above the sleeve K is an oil cup, a cord fastened to C is connected to a weight M and the spring R is attached to the extremity C of the lever by means of a pulley block E. The string $E\ G$ must point to the centre of the pulley when its tension is negligible and the end

of the string is held in the hand, and thus we can easily alter the tightness of the brake. The pulley *G* is not essential, but it enables us to make the adjustment more conveniently.

The automatic adjustment of the brake is effected as formerly by the simultaneous actions of the cord ***M*** and the spring *R*.

Résumé.

We see that in all these automatic brakes there is an error introduced by the tension of the cord, whether it is attached to a weight *M* or to a fixed point. Although this error is small, it cannot be neglected in accurate testing, and hence, when great accuracy is required, we use a simple brake. In comparative tests the error introduced by neglecting the tension of the cord is of little importance.

Beer or Fetu-Deliège Brake (Fig. 163).

The brake band is fixed at *a* and *b* to two iron bars bent to pass over the shaft. *C C* are two iron guide plates so adjusted that their weight balances the brake band about the centre of the shaft.

The rod *c* is a continuation of the brake band, and it is connected to two rollers whose common axis is controlled by the rod *d*, which is attached to a fixed point lower down. These rollers move over the curved edges of the plates *C C*, which are sectors eccentric to the shaft so arranged that when the weight rises the brake is loosened, and when it falls it is tightened.

An iron box round the lower part of the brake

holds the cooling water, and a shield over the brake prevents splashing.

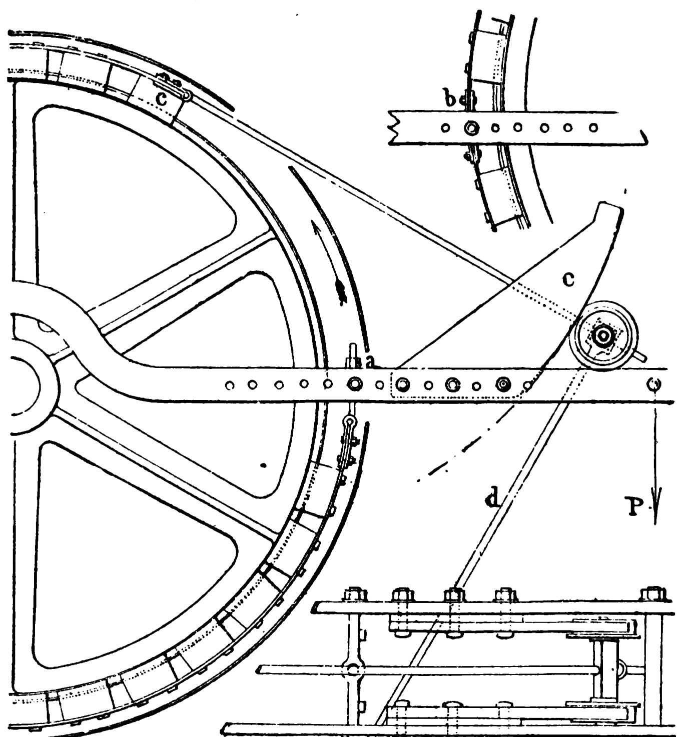


FIG. 163.

Cadiat Brake (Fig. 164).

This brake, which is to a certain extent automatic, has been designed by M. Cadiat, engineer to Mourraille & Cie, of Toulon.

The brake band is adjusted by means of a tension screw on the circumference of a grooved pulley; some of the blocks fit into the grooves, and keep the brake in its place. A cord *a b* passes over a pulley and carries a scale pan, to which is attached a series

of bars (Fig. 164) in such a manner that when the pan rises more of them are raised from the ground, thus increasing the tension, and when it falls back some of them are placed on the ground again, and thus relieve

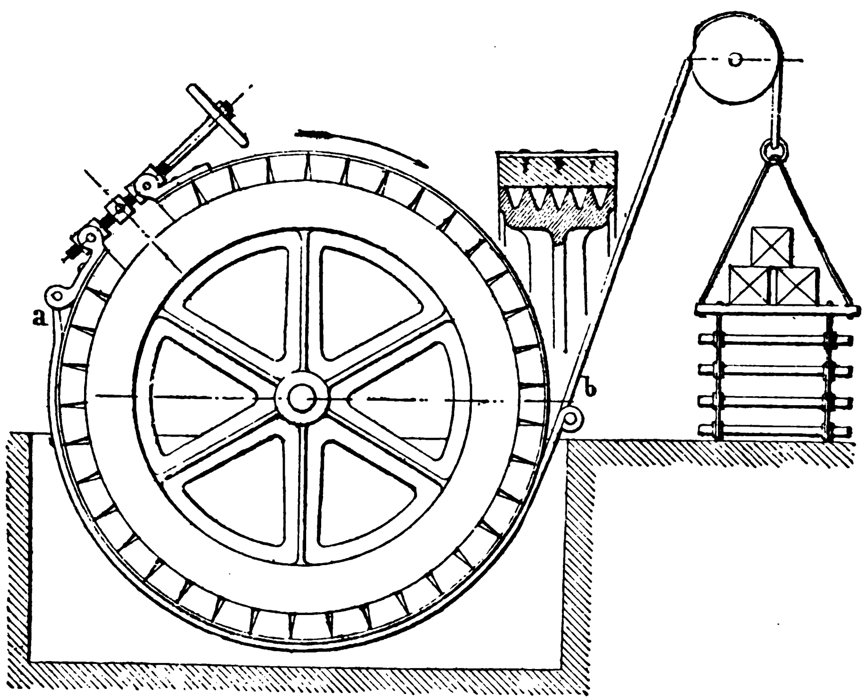


Fig. 164.

the tension. The observer must note the number of bars suspended when making a test; *a* and *b* are two stops which limit the play of the brake band and the lower part of the brake is immersed in soapy water which cools the circumference.

Other Automatic Arrangements.

The arrangement shown in Fig. 165 is a very simple one. One end of the brake band is attached

to the weight P, and the other end to a spring balance q and an additional weight p. The condition of equilibrium is

$$Fr = Pr - (p+q)\ r.$$

Hence the power can be obtained by writing $P-(p+q)$ for P in formula (1).

When the friction increases the weight P tends to

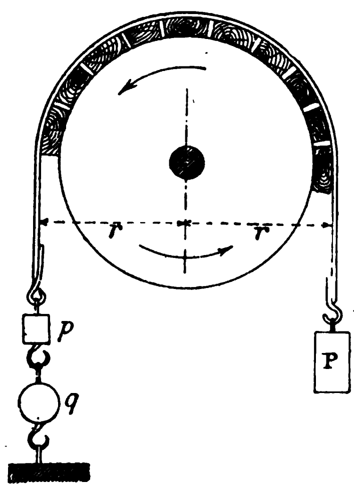


FIG. 165.

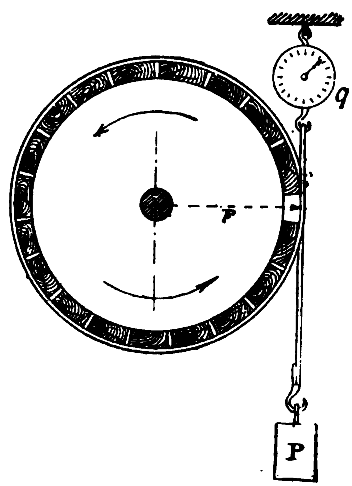


FIG. 166.

rise, its moment Pr, however, remains constant; but since q diminishes, and therefore F increases, the load is increased, and equilibrium is re-established. Conversely when P falls q is increased, and the load diminishes. This arrangement is variously attributed to Navier or to Easton and Anderson.

The arrangement in Fig. 166 is a slight modification of that in Fig. 165. The two extremities of the brake band cross and are situated in the same vertical, and the tension q tends to maintain equilibrium. In

these two arrangements the brake band can be a leather belt, or it can be simply a band of copper with graphite for a lubricant.

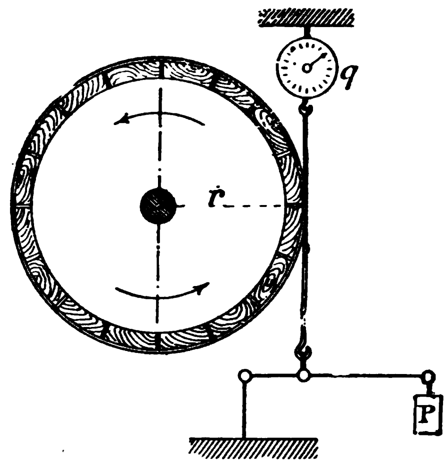


FIG. 167.

In Fig. 167 the tangential pull is measured by a weight P placed at the extremity of a lever or steelyard.

Imray Brake (Fig. 168).

In this arrangement, which is suitable for the measurement of small powers, the arc enveloped by the brake diminishes as the friction increases, and conversely. The brake-band carrying the weight P is connected to a balanced sector A, and carries at its other extremity a weight q.

If the friction increases the weight P rises, but as the arc of the pulley embraced by the band diminishes the friction is diminished, and hence equilibrium is soon established. If P fall, the load is increased, and a balance is soon obtained.

M. Deprez Brake (Fig. 169).

The two levers $e\ c$ and $e\ b$ of the friction blocks are jointed at e and e upon a disc B, and are connected to

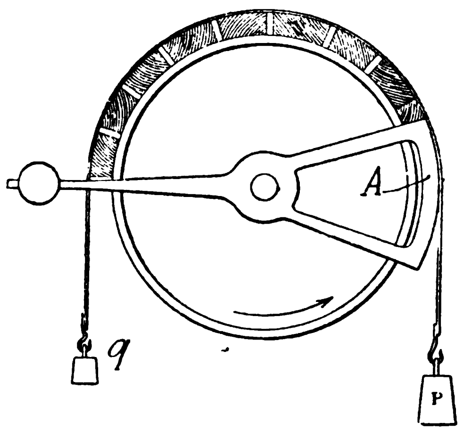


FIG. 168.

a lever $b\ o$, making an angle α with the horizontal. The point O corresponds to the centre of the pulley A which is keyed upon the shaft of the motor. At this point is hung a weight q, which is proportional to the tightness with which we wish the blocks to press on the pulley.

It is evident that the weight Q will produce the maximum force on the blocks when the lever $b\ o$ is horizontal, and will produce no effect at all on them when it is vertical. The tightness is therefore proportional to $\cos \alpha$.

The disc B with the counterweight C is free on the shaft, and carries the weight P, which is to measure the work done.

Suppose now (Fig. 169) that the moment of the

frictional forces upon the pulley A are in equilibrium with the moment $P\,L$. If this friction diminish, the weight P is lowered, and at the same time the angle α diminishes, and hence Q exerts a greater tightening

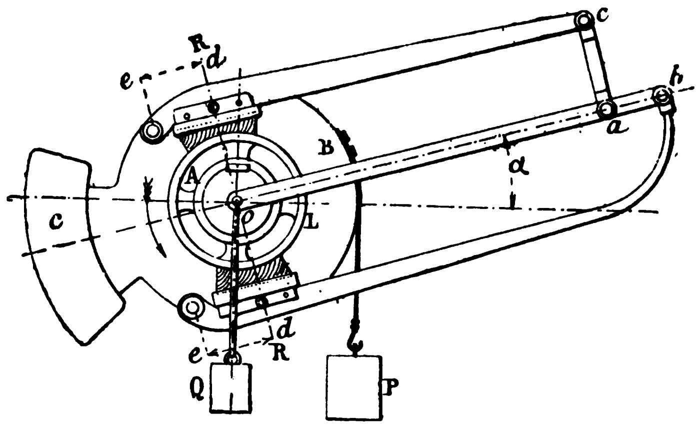


FIG. 169.

force on the friction blocks, which increases the friction, and thus supports the weight P. Reciprocally when the friction increases P is raised; but the brake being slackened, equilibrium soon ensues.

The normal pressures R upon the pulley A are given by—

$$R = Q \cos \alpha \times \frac{a\,o}{a\,b} \times \frac{b\,e}{e\,d} \quad . \; . \; . \; . \quad \text{(lower).}$$

$$R = Q \cos \alpha \times \frac{o\,b}{a\,b} \times \frac{c\,e}{e\,d} \quad . \; . \; . \; . \quad \text{(upper).}$$

These ratios must give equal values of R.

From the practical point of view, the position of the weight Q must produce troublesome side friction, which will prevent smooth running.

We do not believe that this brake has yet been constructed, and it looks rather impracticable.

Carpentier Brake (Fig. 170).

This arrangement requires two pulleys, one (A) keyed on the shaft, and the other one (B) idle. A cord fixed to the idle pulley B, either by a cheek or otherwise, winds itself on that pulley and supports the weight P. On the other side the cord is wound round the fixed pulley, and supports a weight p.

Suppose that the system is in equilibrium, and that the weights P and p do not move; the pulley B is then fixed, and the pulley A turns, rubbing against the cord. For the same weight p, the friction and consequently the tangential effort that the pulley A exerts on the cord a is greater the larger the number of turns, just as in the case of a capstan.

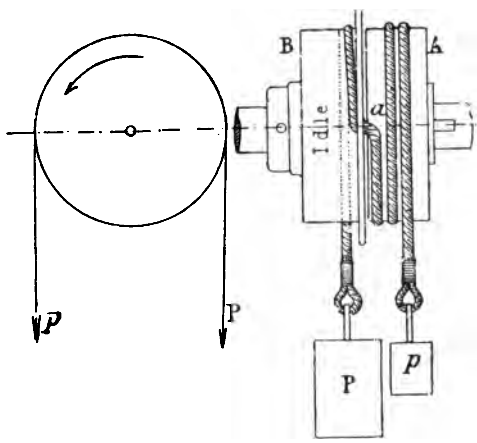


FIG. 170.

If now the friction increases from any cause the pulley B and the weight P will be dragged round in the direction of the motion, and at the same time some of the cord on A will be unwound, thus relieving the friction, and hence equilibrium will soon ensue.

If on the other hand the friction diminish, more cord will be wound on A, thus increasing it.

For a given power it is necessary to adjust the two weights P and p, and to roll a certain amount of cord round A; but once equilibrium is established it is maintained automatically.

It is at once seen that the above arrangement is more complicated than some of the others we have described, and that it is only suitable for small motors.

A still more complicated brake has been constructed on this principle (Fig. 171). It consists of three pulleys A, A' and B. B is placed between the

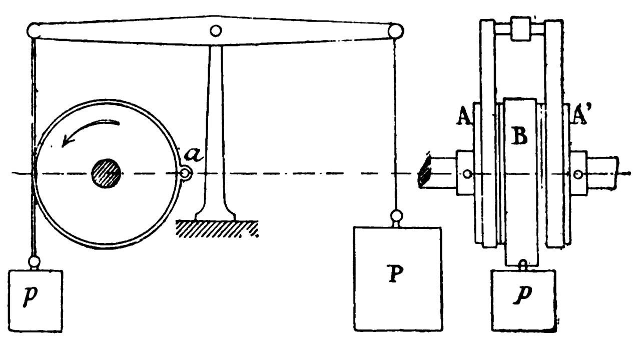


FIG. 171.

others, and is idle, the other two being keyed to the shaft. The cord is replaced by leather straps, and the weight P is carried at the extremity of a lever on the same side as p, but so as to equilibrate it. As this apparatus requires a fourth pulley to be put in connection with the motor, it is more comparable to a dynamometer than to a brake, but it is only suitable for small powers.

This construction seems to us to have little to recommend it, for, besides its complication, it has in practice the drawback of the difficulty of adjusting

the weights p and P, whilst the limits of automatic adjustment are very small.

The arrangement (Fig. 166) is, in our opinion, much superior to those which precede it, firstly because it can be adapted to the pulley of the motor itself, and secondly because it is only necessary to adjust the weight P in order to obtain equilibrium.

Brake and Indicator.

Testing the brake horse power of machines in the workshop presents, as a rule, no great difficulty. It is often even more simply found than the indicated horse power, as the whole test can be made by the foreman fitter, and the final result ascertained more quickly than from the indicator diagram.

In the actual brake test, however, of machines in daily use, difficulties arise. Besides the trouble and expense of setting up the brake we need to disconnect the engine from the shafting which it drives, and the test has to be sufficiently long to enable us to eliminate the errors due to the inertia of the rotating or reciprocating masses. An efficiency test should last for at least one day. Sometimes the brake can only be placed on a transmission shaft. In this case we must calculate the frictional work done between the brake and the machine, and add this on to the work done by the brake.

We see that for a machine in daily use it will often be difficult to apply a suitable brake.

Sometimes the simultaneous use of a brake and an indicator is desirable. It is sufficient to apply the brake only for the short time required to find out the

ratio of the brake to the indicated horse power, and then we can tell what happens subsequently from the diagrams.

When the motor to be tested is coupled to other motors, and the load is variable, then, if it is impossible to disconnect, we must arrange that its load is approximately constant by regulating the admission of the steam and taking off its governor. In this case the other machines have to supply more or less power depending on the load. The diagrams of the indicator will give the I.H.P. of the engine being tested.

CHAPTER IX.

THE DYNAMO USED AS A BRAKE.

Magnetic Brake.

THE dynamo can sometimes be conveniently used as a brake. It is driven by the machine the brake power of which we want to measure, and the electric power it generates is expended in heating suitable resistances. If we measure the total electromotive force E of the dynamo by a suitable voltmeter and the current I by an ammeter, then the total electric power generated is $E\ I$ watts or $\frac{E\ I}{746}$ horse power. If the commercial efficiency of the dynamo at this load is η, then the mechanical power given to it by the machine is

$$\frac{E\ I}{746\ \eta}$$

and this is the brake horse power wanted.

Dynamometer.

The dynamo being a reversible machine can conversely transform electrical power into mechanical power, and can be used to drive workshop machines or tools; used in this manner it is called a motor. Multiplying the electrical power given to the motor by η we at once deduce the power given out by the motor.

We see, then, that the dynamo is also a transmission dynamometer.

Use as a Brake.

We couple a dynamo of sufficient power to the engine whose brake horse power has to be determined. The most suitable dynamo is one which is shunt wound, giving a practically constant pressure at constant speed. We arrange upon the engine shaft a suitable pulley, so that the dynamo runs at its proper speed when the engine speed is normal.

The connections are shown in Fig. 172.

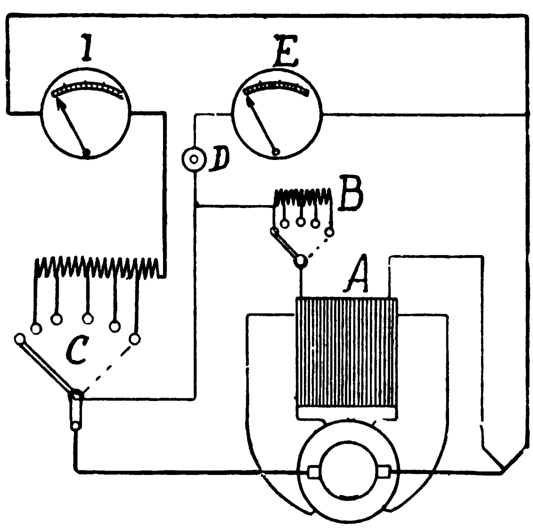


FIG. 172.

A is a shunt dynamo.
B is the field adjusting rheostat.
C resistances to absorb the load.
D voltmeter key.
E voltmeter.
I ammeter.

The shunt current of the dynamo can be regulated by the hand switch at *B*, so as to obtain the proper voltage. We start the machine on open circuit, and when it attains its voltage we close the external circuit and regulate the load until the desired horse power is obtained.

THE DYNAMO USED AS A BRAKE

Calculation of the Work.

The reading of the ammeter gives us the current I, and the voltmeter gives us the voltage E.

Let η be the efficiency of the dynamo at the load $E\,I$ watts,

T = the brake horse power which has to be measured,

then $\eta\,T$ = electric power,

$$= \frac{E\,I}{746} \text{ horse power.}$$

Therefore $T = \dfrac{E\,I}{746\,\eta}$ horse power.

Direct Measurement of η.

We have first of all to determine the efficiency of the dynamo at various loads. These efficiencies can generally be had from the maker of the machine, but it is always preferable to find them ourselves. They can be obtained by calculation or by several experimental methods, all of which lead to the same result. A particularly convenient method is the one first employed by Mr. James Swinburne.

The dynamo is a reversible machine, and so its efficiency as a motor is practically the same as its efficiency as a dynamo.

Efficiency as a Dynamo.

If we send a current I through the armature of a motor and E be the potential difference at its terminals, then the power given to it is $\dfrac{E\,I}{746}$. If now we apply a friction brake to a pulley on its shaft, and

if T be the measured brake power absorbed by the frictional forces—

$$T = \eta \frac{E\,I}{746}; \quad \eta = \frac{746\ T}{E\ I}.$$

Efficiency Curve (Fig. 173).

If we measure kilowatts $\frac{E\ I}{1000}$ horizontally and efficiencies (η) vertically, we obtain the efficiency curve (Fig. 173), and this curve remains practically always constant.

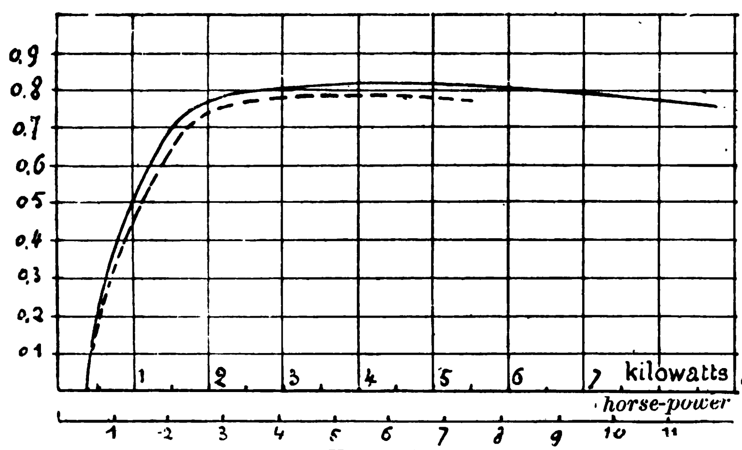


FIG. 173.

So long as the load does not pass certain values the efficiency of machines of certain sizes is much the same. The following table of approximate numbers will be found useful :—

Horse Power	1	2	3	5	10	50	100
Efficiency	0·70	0·78	0·81	0·82	0·83	0·90	0·92

Above 100 horse power the efficiencies attain values of 95 or even 97 per cent.

CHAPTER IX. *A*.

STEAM TURBINES.

Hypothetical Equivalent Indicated Horse Power.

FOR steam turbines there is nothing analogous to the ordinary indicated horse power diagrams which we get for reciprocating engines. The individual pressures on the many thousands of small blades cannot be ascertained, although the resulting power delivered by the engine to its shaft may be known and the amount of external work performed by it. For instance, when the turbine drives a dynamo, the electrical horse power generated in the latter can be accurately measured by instruments of precision. The efficiency of a dynamo can also be found by any of the various standard electrical methods, and hence the brake horse power of the turbine is known. The accuracy of this determination is probably greater than in the case of the indicated horse power of an ordinary steam engine.

It is customary to assume that in an ordinary steam reciprocating engine of good design the brake horse power is 86 per cent. of the indicated horse power (see Chapter VII.). This agrees closely with the mean of the results of many published tests of ordinary steam engines. For the sake of comparison we assume that in turbines the ratio of the brake to the indicated horse powers is also 0·86. Hence, knowing the brake horse power, we find the indicated. The

horse power found in this manner is called the *hypothetical equivalent indicated horse power.*

Marine Steam Turbines.

A similar method is followed in marine steam turbines.

The resistance of the ship is calculated, and hence the "propulsive horse power" required to drive it at a given speed can be found. For vessels with ordinary reciprocating engines it is generally found that the propulsive horse power is about 55 per cent. of the indicated horse power. In some torpedo boats the ratio of the propulsive to the indicated horse powers may be as high as 0·6 and the lowest value is about 0·4. For cross-channel boats, liners, cruisers and destroyers we may take 0·55 as the ratio.

The determination of the propulsive horse power has been made possible by the work of the late Mr. William Froude. He determined the resistance to passage through the water of a model of a ship in his testing tank, and then from his experimental results calculated what the resistance of the ship would be.

Knowing then the propulsive horse power and its ratio to the indicated horse power, we find the hypothetical equivalent indicated horse power of a marine steam turbine.

It will be seen that the testing of a steam turbine is much simpler than the testing of an ordinary steam engine. It is also much less liable to get out of order. The only effect of bad priming on the part of the boiler is to make it turn a very little slower, whilst with a steam engine there is a great risk of the cover of the cylinder being blown off.

CHAPTER X.

PROPERTIES OF STEAM.

Equivalence of Work and Heat.

WHEN heat by its action does mechanical work, the quantity of heat that disappears is always exactly proportional to the work done, and conversely.

We conclude from the above that heat and energy are measured in the same units and are mutually convertible. The mechanical equivalent of heat is the amount of work required to raise the temperature of one pound of water one degree Fahrenheit. It is nearly 778 foot-pounds. Conversely we could speak of the thermal equivalent of work. One foot-pound is equal to the $\frac{1}{778}$th part of a British Thermal Unit; i.e. the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit (B.T.U.).

The law of the equivalence of heat and work is independent of the nature and the constitution of the body; like the law of universal gravitation it has been deduced from observation and experiment.

Saturated Vapour. (See the tables in the Appendix.)

Consider a pound of water contained in a cylinder whose sides are non-conductors of heat and the bottom part allowing heat to come through from a furnace.

At a pressure of one atmosphere the water will boil at 212° F. If, however, we suppose that a weightless piston exerts upon the water a pressure p greater than the atmospheric pressure, then the water will heat above 212° F.

So long as the pressure p remain constant the temperature t does not change, however hot the furnace may be. The greater heat only makes it boil more rapidly.

The vapour formed in the presence of water is called saturated vapour; that is, its pressure and density are the greatest possible corresponding to the temperature t of the boiling water.

Reciprocally the pressure p of a saturated vapour depends only on the temperature t, and not on the volume which it occupies. Regnault has constructed very elaborate tables of pressures and temperatures which embody the results of his experimental researches. The tables we give in the Appendix have been calculated by Zeuner for every tenth of an atmosphere from Regnault's tables.

The density d of water vapour is equal to 0·622 of that of air at the same temperature and pressure. The weight of one cubic foot in pounds and the volume in cubic feet of one pound are given in the Appendix at the temperature of 39° F., which is the temperature at which water has its maximum density. The mass of a cubic foot of water is 62·4 pounds nearly. The density d of water vapour or the mass in pounds of one cubic foot is equal to 0·622 times that of air at the same temperature and pressure.

The mass of one cubic foot is d pounds and the

volume of one pound is $\frac{1}{d}$ cubic feet. These quantities are tabulated in the Appendix.

At the temperature of 39° F. the volume of one pound of water is 0·016 of a cubic foot nearly, and the volume of one pound of water vapour at that temperature is 20 cubic feet. Hence the ratio of the volume of water vapour to the volume of water that produces it at 39° F. is $\frac{20}{0 \cdot 016}$, i.e. 1250. At 212° F. this ratio is about 1650.

Total Heat of Vaporization, U.

This includes both the heat required to raise one pound of water from 32° F. to t° F., and the latent heat of steam at t° F. Regnault gives the following formula:—

$$U = 1082 + 0 \cdot 305\, t.$$

This quantity of heat increases with t. It is also the heat that leaves one pound of vapour in cooling down from t° F. to 32° F.

Latent Heat, L.

The heat given to a pound of water in order to turn it into vapour without altering its temperature is called the latent heat of steam. If L be the latent heat at t° F., then—

$$L = 1114 - 0 \cdot 695\, t.$$

We see that the latent heat L diminishes as the pressure and consequently the temperature rises.

Superheated Steam.

After the complete vaporization of a pound of water, if we continue the heating the steam becomes super-

heated. The greater the superheating the more nearly does the vapour when it expands obey Boyle's law.

The specific heat of water vapour being 0·475 (or more simply 0·48) the heat required to superheat a pound of water from t'°F. to t°F. is

$$0{\cdot}475\ (t-t'),$$

and the total heat contained in one pound of the vapour is

$$1082 + 0{\cdot}305t' + 0{\cdot}475\ (t-t').$$

This is also the total heat that would be given up by a pound of superheated steam in cooling from t to 32° F.

Saturated Steam. Compression.

If we have saturated steam confined in a cylinder by means of a piston, and if we compress the piston, the heat due to the compression will more than compensate for the diminished volume, and hence the steam will become superheated. All vapours do not behave in this manner. Alcohol vapour, for example, is condensed by compression.

Saturated Steam. Adiabatic Expansion.

When the piston is allowed to expand, no heat being supposed to enter or leave the cylinder during the expansion, then the increased volume fails to compensate for the diminished pressure, and so some of the steam condenses and furnishes to the remaining steam the heat required to keep it in the form of vapour during the expansion. Towards the end of the stroke some of the condensed steam may be re-evaporated again.

Expansion of a given Quantity of Vapour.

If during the expansion we supply the vapour with the heat required to maintain it in the gaseous form, the mass of the vapour will remain constant. In this case if p be the pressure in pounds per square inch and v be the volume in cubic feet, the approximate law of the expansion will be

$$p\ v^{1\cdot 05} = \text{constant}.$$

Boyle's or Mariotte's Law.

If more heat be given to the vapour during the expansion than that required to keep it gaseous, it becomes superheated and it obeys Boyle's law very approximately, which may be given as follows :—*The volume of a mass of gas varies inversely as the pressure.*

Let V_o = the initial volume of the steam.

V = the final volume or the volume of the cylinder.

P = the initial pressure.

P' = the final pressure.

We have

$$\frac{P}{P'} = \frac{V}{V_o} \text{ or } P\ V_o = P'V = \text{constant}.$$

If we plot out a curve having volumes for abscissae and pressures for ordinates we get a hyperbola.

Calculation of the Mean Pressure p_m.

We can calculate the mean pressure p_m of a theoretical diagram (Fig. 174) when we know the ratio of the expansion $\frac{V}{V_o}$. A knowledge of p_m is sometimes

useful when we are designing an installation or when we are calculating the indicated horse power of an engine to which an indicator cannot be applied. The

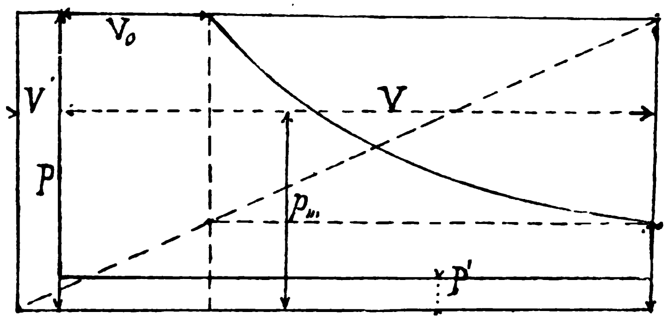


FIG. 174.

total work of the steam during a stroke of the piston is p_m V, and it is equal to the sum of the following :—

(1) The work during admission $P\ V_o$.

(2) The work during expansion. Assuming Boyle's law, this equals $P\ V_o \log_e \frac{V}{V_o}$

(3) The negative work done during the return stroke $P'V$.

Summing up these three works and dividing by V, we get

$$p_m = P\frac{V_o}{V}\left(1 + \log_e \frac{V}{V_o}\right) - P'.$$

The following table gives the hyperbolic logarithms of the expansion ratio $\frac{V}{V_o}$:—

Hyperbolic Logarithms.

$\frac{V}{V_0}$	log. hyp.	$\frac{V}{V_0}$	log. hyp.	$\frac{V}{V_0}$	log. hyp.	$\frac{V}{V_0}$	log. hyp.
1·4	0·3365	3·7	1·3083	6·0	1·7918	8·3	2·1163
1·5	0·4055	3·8	1·3350	6·1	1·8083	8·4	2·1282
1·6	0·4700	3·9	1·3610	6·2	1·8245	8·5	2·1401
1·7	0·5306	4·0	1·3863	6·3	1·8405	8·6	2·1518
1·8	0·5878	4·1	1·4110	6·4	1·8563	8·7	2·1633
1·9	0·6419	4·2	1·4351	6·5	1·8718	8·8	2·1748
2·0	0·6931	4·3	1·4586	6·6	1·8871	8·9	2·1861
2·1	0·7419	4·4	1·4816	6·7	1·9021	9·0	2·1972
2·2	0·7885	4·5	1·5041	6·8	1·9169	9·1	2·2083
2·3	0·8329	4·6	1·5261	6·9	1·9315	9·2	2·2192
2·4	0·8755	4·7	1·5476	7·0	1·9459	9·3	2·2300
2·5	0·9163	4·8	1·5686	7·1	1·9600	9·4	2·2407
2·6	0·9555	4·9	1·5892	7·2	1·9741	9·5	2·2513
2·7	0·9933	5·0	1·6094	7·3	1·9879	9·6	2·2618
2·8	1·0296	5·1	1·6292	7·4	2·0015	9·7	2·2721
2·9	1·0647	5·2	1·6487	7·5	2·0149	9·8	2·2824
3·0	1·0986	5·3	1·6677	7·6	2·0281	9·9	2·2925
3·1	1·1314	5·4	1·6864	7·7	2·0412	10	2·3026
3·2	1·1632	5·5	1·7047	7·8	2·0541	11	2·3979
3·3	1·1939	5·6	1·7228	7·9	2·0669	12	2·4849
3·4	1·2238	5·7	1·7405	8·0	2·0794	13	2·5649
3·5	1·2528	5·8	1·7579	8·1	2·0919	14	2·6391
3·6	1·2809	5·9	1·7750	8·2	2·1041	15	2·7081

The pressure P in the cylinder is always less than the pressure in the boiler. It is sometimes obtained directly by a manometer placed on the valve chest.

The back pressure, or the pressure during exhaust, is 15·6 to 17 pounds per square inch in non-condensing engines and 2·2 to 3 pounds in condensing engines.

In the preceding calculation of the mean theoretical pressure p_m we have not taken into account the volume of the steam in the clearance space. This modifies the pressure during the expansion, as the volume that expands is in reality $V_0 + V'$. If there is compression during the return stroke (cushioning), we have again to take the clearance space into account.

Suppose that $V_{\circ} = m\ V$ and $V' = m'\ V$, then we can show that

$$p_m = P \left\{ m + (m + m') \log_e \frac{1 + m'}{m + m'} \right\} - P''$$

or $\quad p_m = K\ P - P''.$

Where $K = m + (m + m') \log_e \dfrac{1 + m'}{m + m'}$.

We have calculated the values of K for a clearance m' equal to $\frac{1}{20}$ and the results are given in the following table :—

m	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{7}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$	$\frac{6}{10}$	$\frac{7}{10}$	$\frac{8}{10}$
K	0·39	0·44	0·47	0·50	0·55	0·62	0·72	0·85	0·90	0·95	0·98

The pressure calculated by this formula goes on the assumption that there is no lap or lead, and also it does not take account of re-evaporation.

Theoretical Weight of Vapour per Horse Power Hour.

The weight of vapour per horse power hour may be calculated from the mechanical theory of heat. We will not reproduce here the calculation which will be found in treatises on Thermodynamics, as it has no direct practical application. It is sufficient to indicate the results so that they may be compared with the weights of vapour calculated from the diagram.

Absolute Pressure in Atmospheres.	1·5	3	4	5	6	8	10
Pounds per Horse Power Hour Condensing.	15·4	12·7	11·8	11·3	10·9	10·3	9·9
Pounds per Horse Power Hour Non-Condensing.	72·7	33·1	25·4	22·9	20·7	18·1	16·5

Weight of Dry Steam per Horse Power Hour. Warrington's Diagram.

We have seen that the indicated work is given by the formula—

$$\text{H.P.} = \frac{2\,p\,L\,A\,N}{33000}$$

$$\text{or } 33000 \text{ H.P.} = 144 \times p \times 2\,L.A.N.$$

when the area of the piston is expressed in square feet. When H.P. is 1 and p is 1 the work per hour is

$$33000 \times 60 = 144 \times (2\,L\,A\,N \times 60)$$

Hence if we call the volume swept out by the piston in one hour V, we have

$$V = 2\,L.A.N. \times 60$$

$$\text{Therefore } V = 13750 \text{ cubic feet.}$$

If the pressure p be expressed in atmospheres, then since one atmosphere is 14·69 lbs. per square inch

$$V = \frac{13750}{14{\cdot}69}$$

$$= 935{\cdot}4 \text{ cubic feet.}$$

Hence at a pressure of one pound per square inch the volume of the fluid used per horse power hour is 13,750 cubic feet, and when the pressure is one atmosphere it is 935·4 cubic feet.

This volume will vary inversely as the pressure, so that we have

$$V = \frac{13750}{p'_m} \text{ when } p'_m \text{ is in pounds.}$$

$$\text{or } V = \frac{935{\cdot}4}{p_m} \text{ when } p_m \text{ is in atmos.}$$

If V_e be the volume of unit mass of the steam at the

final pressure (p_e) got from the indicator diagram, then the mass M of the steam will be given by

$$M = \frac{V}{V_e}$$

$$= \frac{935{\cdot}4}{p_m \, V_e}$$

The value of V_e for various values of p_e is tabulated in the Appendix. To find the expenditure of dry steam, it is only necessary to divide p_m into the value of $\frac{935{\cdot}4}{V_e}$, which can be obtained from the last column of the table. The final pressure of the steam in the cylinder (p_e) is obtained from the indicator diagram.

Corrections.

The clearance spaces increase the steam consumption, but compression tends to diminish it.

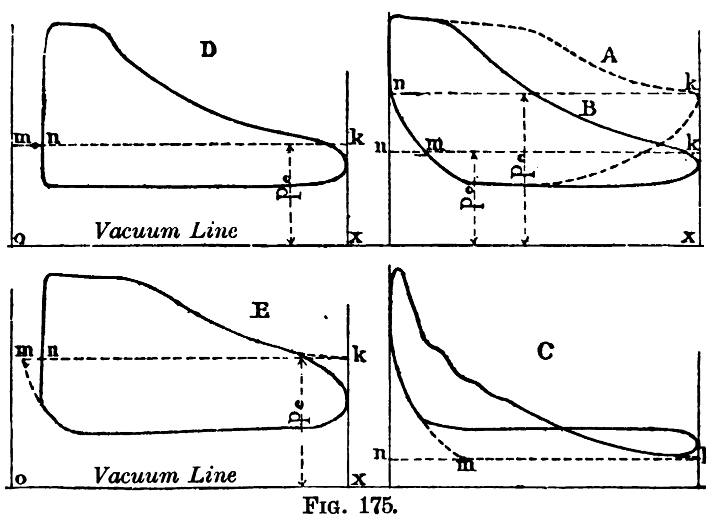


FIG. 175.

To take account of this action draw through the point K which determines the final pressure p_e (Fig. 175) the line $k\ m$ parallel to $o\ x$. This

line will cut in m the curve of compression prolonged if necessary. The corrected mass of the vapour M_1 is given by

$$M_1 = M \frac{km}{kn}$$

In the curve A (Fig. 175) it will be seen that the final value of the back pressure during the compression is p_e. The effect of the clearance space is therefore annulled. In this case we have

$$\frac{km}{kn} = 1$$

In B and in C $\frac{km}{kn}$ is less than unity, and in D and E it is greater. This ratio is a maximum in D, where the compression is zero.

Example.

In the diagram (Fig. 107) we have—

$$p_e = 11{\cdot}4 \text{ and } p_m = 17 \text{ lbs. per square inch.}$$
$$= 1{\cdot}2 \text{ atmospheres.}$$

From the tables in the Appendix we find that

$$V_e = 33{\cdot}7 \text{ and } \frac{935{\cdot}4}{V_e} = 27{\cdot}7$$

The mass of the vapour will therefore be $\frac{27{\cdot}7}{1{\cdot}2}$ lbs.

i.e. 23·1 lbs.

$$\frac{km}{kn} \text{ found from the diagram } = \frac{89}{95}$$
$$= 0{\cdot}94$$

The corrected mass will therefore be $23{\cdot}1 \times 0{\cdot}94$
$$= 21{\cdot}7$$

This mass, however, is not quite exact, as the steam is always more or less wet.

For accurate tests we can measure the mass of the wet steam consumed by the method of direct measurement which we shall soon discuss.

Application to Machines with Two Cylinders.

The preceding method applies also to engines with two cylinders, Woolf or compound. The pressure p_e at the end of the expansion is given by the diagram of the low pressure cylinder. It is necessary, however, to calculate the mean pressure p_m common to the two cylinders.

Suppose that we have determined the mean pressure p'_m in the high pressure cylinder, whose volume is V, and p''_m the mean pressure in the low pressure cylinder, whose volume is V. Then we may write

$$p_m = p'_m \frac{v}{V} + p''_m$$

The correction can be made by finding the point m on the curve of compression of the low pressure cylinder.

Example.

To calculate the consumption of steam from the diagrams in Figures 110 and 111.

The diagram A gives us $p'_m = 31{\cdot}9$ lbs. per square inch. The diagram B gives us $p''_m = 21{\cdot}3$ lbs. per square inch; these being the mean of the pressures on the two sides of the cylinder.

The ratio of the volumes $\frac{v}{V} = \frac{1}{2{\cdot}84}$

Hence the mean pressure will be

$$p_m = \frac{31{\cdot}9}{2{\cdot}84} + 21{\cdot}3 = 11{\cdot}2 + 21{\cdot}3$$
$$= 32{\cdot}5 = 2{\cdot}3 \text{ atmos.}$$

On prolonging the curve B we find that $p_e = 14{\cdot}2$.

The tables give us $V_e = 28{\cdot}1$ and $\dfrac{935{\cdot}4}{V_e} = 34{\cdot}8$

$$\text{Hence } \frac{34{\cdot}8}{p_m} = \frac{34{\cdot}8}{2{\cdot}3} = 15{\cdot}2 \text{ lbs.}$$

$$\text{Correcting for compression, } 15{\cdot}2 \times \frac{88}{90} = 15 \text{ lbs.}$$

Making an allowance of ten per cent. for the increased consumption due to the wetness of the steam, we find that the consumption is 16·5 lbs. per h.p. hour. By actual measurement the consumption was found to be 16·6 lbs.

Direct Measurement.

The consumption can be got directly by measuring the amount of water injected into the boiler.

For small engines, especially when the boiler furnishes steam for other purposes, we can measure the consumption of wet steam by passing the exhaust steam into a surface condenser. The mass of water collected is the mass of wet steam consumed by the engine.

Wet Steam. Priming.

In what precedes we have supposed that the steam is dry saturated vapour; but in practice this assumption is not permissible, as the steam coming from the boiler contains (except when superheated) a certain proportion of water. This result is due either to priming, or to a bad arrangement of the steam pipe of the boiler.

Priming.

When the ebullition of the water in the boiler is very violent the steam carries over with it to the

cylinder a certain amount of water, and priming is said to take place. If the orifice of the steam pipe be too near the surface of the water in the boiler the priming will be increased.

This wet vapour causes a loss of heat, because it is only the latent heat of the vapour that does work. In addition, the water getting over may cause the piston to rupture the end of the cylinder. We can find out approximately the relative humidity of the steam by the appearance of a small jet of steam supplied by means of a cock in the dome of the boiler. We must not take this jet by means of a pipe to any great distance from the dome, as in passing along this pipe some of it would be condensed.

So long as the jet of steam issuing from the cock is transparent or slightly grey, the vapour is dry; if the vapour presents a whitish appearance it contains two to three per cent. of water; but it is still considered dry steam.

If the issuing vapour looks misty, then it is called wet steam, and the proportion of water is determined as follows:—

Measurement of the Water Carried Over.

Various methods of doing this have been proposed. We shall confine ourselves to describing two of them, namely: (1) the calorimetric or condensing method, and (2) the method of dissolving sea salt.

First Method.

We fix upon the steam pipe, close to the dome of the boiler, a cylindrical tube with a stopcock which has a flexible tube fastened to its nozzle. We heat

the tube by allowing a certain amount of vapour to pass through it. Then we plunge the flexible tube into a suitable receptacle which is protected from loss of heat, and contains a mass M of water and a thermometer. This receptacle, or calorimeter, as it is generally called, can be gauged and calibrated, or, more simply, it may be placed on the scale pan of a balance. We allow the steam to condense in the water, but shut off the cock before the water attains 212 deg. F.

Let M = the initial mass of the water in the calorimeter.

t = the temperature of the water.

a = the proportion of dry steam.

b = the proportion of water.

$m = a + b$, the increase of the mass of water.

t' = the temperature of the steam in the boiler.

t'' = the temperature of the water in the calorimeter at the conclusion of the experiment.

L = the latent heat of steam at t' deg. F.

Then since the number of units of heat gained by the water will equal the number lost by the wet steam, we have:

$$a\ (L + t' - t'') + b\ (t' - t'') = M\ (t'' - t')$$
$$\text{and } a + b = m.$$

From these two equations the percentage quantity of water in the steam can easily be found.

Second Method.

In the second method we saturate the water in the boiler to a known degree by means of salt (n

ounces per gallon of water, for example). Now dry steam carries over no salt to the cylinder, and so all the salt that comes over has been taken by the priming water. An analysis of the water got from the drain cocks of the cylinder will show us the amount of priming that is taking place.

Condensation of Vapour.

When vapour is in contact with cold water some of it will condense, and the quantity of heat lost by the vapour in condensing will be nearly all given to the water. A negligible amount will be given to the sides of the containing vessel and taken for re-evaporation.

Let M = the mass of vapour which has to be condensed.

M' = the mass of water required for this purpose.

t = the temperature of this water (50° to 70° F.).

t' = the temperature of the mixture after condensation.

L = the latent heat of steam at the exhaust pressure.

Then after the condensation we shall have:

$$M(L-t')=M'(t'-t).$$

Hence
$$M'=\frac{M(L-t')}{t'-t}.$$

For example, when the exhaust pressure is two atmos. $L=941$. Suppose that $t=60°$ F., $t'=100°$ F., then

$$M'=\frac{941-100}{40}M=21\ M.$$

Hence we require twenty-one pounds of water for every pound of steam condensed.

CHAPTER XI

§ 1. Vaporization.

FUELS.

Calorific Power.

THE remark that fuels store up for us the heat of the sun, the source of all vegetation, has been attributed to G. Stephenson.

Every body that is capable of combining with the oxygen in the air is said to be combustible. The combination is called the combustion, and during the process the heat and light are restored.

The combustible materials of commerce contain (1) a certain quantity of oxygen and hydrogen combined in the proportion of 8 to 1, which forms water and does not furnish heat; and (2) inert substances which do not give heat, such as nitrogen, the mineral matters which form the ash, pyrites or disulphide of iron which we can neglect on account of the feeble calorific power of sulphur; and (3) carbon and free hydrogen which are the elements whose combustion produces the heat.

The quality of a fuel is judged from its calorific power, its density, its cohesion, its appearance in the fire, and from the nature of the ash.

The ash of bituminous fuels is apt to form a clinker which obstructs and burns the grate, whilst the ash

of earthy fuels falls without obstructing the free passage of air.

The *calorific power* P of a fuel is the number of British thermal units which a pound of it gives up on burning. According to Dulong it is equal to the sum of the calorific values of its elements, carbon and free hydrogen. Favre and Silbermann found that the calorific value of carbon C was equal to 14,545, and that of hydrogen was 62,028.

In the analysis of fuels we take the oxygen and nitrogen together, as the latter is always of little importance. Since the constitution of water is eight parts of oxygen O to one of hydrogen H, we see that the proportion of free hydrogen will be

$$H-\frac{O}{8}$$

The formula of Dulong for pure combustion is

$$P=C\times 14545+\left(H-\frac{O}{8}\right)62028$$

This law is not very exact for very dense bodies which have a large proportion of hydrogen, but their calorific powers have been found directly by experiment.

Let a = the fractional quantity of water in the fuel,
and b = the fractional quantity of ash produced.

The true calorific value P' of the fuel will then be found from

$$P'=P\,(1-a-b).$$

We have made the assumption (1) that the water vapour formed during the combustion is condensed to 32 deg., the total heat of vaporization being re-

stored; and (2) that the ashes formed by the combustion are also cooled down to 32 deg.

In the heating of locomotive boilers these assumptions are not allowable. The steam which passes up the chimney takes with it 1,147 B.T.U. per pound, together with the heat $0{\cdot}5\ (t-212)$ where t is the temperature of the gases in the chimney.

If h be the average mass of the hydrogen contained in one pound of fuel, its combustion gives a mass of water $9\ h$, and the total quantity of water vaporized will be $a+9\ h$.

It is further necessary to deduct from P' the heat which remains in the ashes and clinker, but these losses, together with that due to the temperature t of the gases in the chimney, are included in the losses due to the inefficiency of the furnace.

The efficiency is obtained by finding the number of absolute units of heat generated by the combustion of the fuel. Hence we use the formula—

$$P'=P\ (1-a-b)\ -\ 1114\ (a+9\ h).$$

Commercial fuels may be divided into five classes—

(1)	Wood,	charcoal.
(2)	Peat,	turf.
(3)	Lignites,	brown coal.
(4)	Coal,	coke.
(5)	Anthracite.	

Wood.

We class woods commercially into heart-woods and sap-woods. Woods for heating purposes are divided into new wood, drift wood, and wood with the bark removed.

Brisson gives the following values for the specific gravities of woods:—

Heart-oak . .	1·17	Lime . . .	0·60
Beech, Ash .	0·85	Willow . .	0·58
Alder, Apple .	0·80	Fir (male) .	0·55
Maple, Cherry	0·75	„ (female)	0·49
Elm, Walnut .	0·67	Poplar . .	0·38
Pear	0·66	Cork . . .	0·24

Wood always contains a certain proportion of water. It is only employed after being partially dried in the air, or dried in ovens at a temperature of about 300° F. The felling of trees is done during the winter. They then contain from forty to forty-five per cent. of water; after six months they contain twenty-six per cent., after a year twenty per cent., and after eighteen months, seventeen per cent. However long they are exposed they always contain about fifteen per cent. of water. Woods which have been completely dried in hot air ovens will, if left in the open air, gradually absorb moisture until they contain from fourteen to sixteen per cent. of water.

Woods which have been dried at a temperature of about 280° F., contain approximately 0·5 of carbon, 0·01 of free hydrogen, 0·46 of oxygen and hydrogen in the ratio necessary to form water, 0·01 of nitrogen, and 0·02 of ashes.

By Dulong's law,

$$P = 0{\cdot}5 \times 14500 + 0{\cdot}01 \times 62000$$
$$= 7870 \text{ B.T.U. approximately.}$$

Applying the required corrections,

$$P' = 7870 - 1114\ (0{\cdot}46 + 0{\cdot}09).$$
$$= 7257 \text{ B.T.U.}$$

Suppose the wood contains thirty per cent. of water,

$$P' = 7257 \times 0{\cdot}7 - 1114 \times 0{\cdot}3.$$
$$= 5746 \text{ B.T.U.}$$

The table (A) gives the results of the experiments of M. Chevandier on the calorific power per cubic yard of various kinds of woods dried at 280° F. The calorific power of one pound of fuel varies between 7600 and 8000, and is approximately equal to the number 7870 found above.

A.—*Experimental Results by M. Chevandier.*

	Nature of Wood.	Weight in lbs. per cubic yard.	Carbon.	Free Hydrogen.	Calorific Power.	
					Per cubic yard.	Rel.
Quarterings.	Oak	607	315	4·37	4,900,000	1
	Beech ...	607	312	4·42	4,850,000	0·994
	Hornbeam ...	604	300	3·8	4,650,000	0·95
	White Oak...	600	298	4·15	4,620,000	0·945
	Birch ...	565	288	6·1	4,580,000	0·939
	Alder ..	490	250	4·95	3,970,000	0·812
	Fir	465	237	4·37	3,720,000	0·762
	Pine... ...	430	235	2·98	3,450,000	0·706
Bundles of branches.	Beech ...	501	250	3·55	3,900,000	0·795
	Fir	480	245	4·52	3,860,000	0·79
	Pine ...	470	240	4·37	3,800,000	0·779
	Hornbeam ...	597	242	3·08	3,740,000	0·764
	Birch ...	450	228	4·85	3,650,000	0·747

The above weights are calculated on the supposition of sixty per cent. of solid wood to the cubic yard. A cord of wood has a volume of four and three-quarter cubic yards very nearly.

Heart-woods burn at the surface, producing a large quantity of carbon, whilst sap-woods split in the fire and burn violently to the centre, giving out flames until they are all consumed.

The more finely divided the wood is, the more rapid

the combustion and the higher the efficiency, because the air is better utilized; but splitting up the wood is expensive.

Wood Carbon.

This is obtained by carbonizing the wood in stacks, and the return varies with the temperature as follows:—

From 300° to 500°, return = 37 to 40 %.
From 540° to 650°, return = 32 to 36 %.
(red charcoal).
From 670° to 780°, return = 18 to 31 %.
(black charcoal).
From 780° to 2,400°, return = 17 to 18 %.
(charcoal is hard and black).

The return, which is found by weighing, varies for black charcoal from eighteen to twenty per cent.

The weight of a cubic foot varies according to the wood it is made from :—

For oak and beech . . .	16 to 17 lbs.
For birch	15 to 16 lbs.
For pine	14 to 15 lbs.

According to Ebelmen the mean composition of dry carbon is as follows :—

Carbon	0·875.
Hydrogen	0·03.
Oxygen and Nitrogen	0·075.
Ash	0·02.

The free hydrogen is nearly 0·02, and hence—

$$P = 0{\cdot}875 \times 14500 + 0{\cdot}02 \times 62000.$$
$$= 13930 \text{ B.T.U.}$$

Hence, making allowance for the ash, we get for the pure carbon 14200.

For a carbon containing 6 per cent. of water and 4 per cent. of ash we have:

$$P' = 14200 \times 0{\cdot}9 - 1114\ (0{\cdot}06 + 9 \times 0{\cdot}04).$$
$$= 12780 - 1114 \times 0{\cdot}42.$$
$$= 12300.$$

The relative values of carbons are as their weights per cubic foot.

Tan Bark. Sawdust.

These combustibles are burned in special furnaces. Theoretically they have the same calorific value as the fuels from which they are derived. Tan bark, after it is pressed, contains about 48 per cent. of water, 8 to 12 per cent. of ash, and weighs about 52 lbs. per cubic foot. Its true calorific value deduced from that of dry wood containing 2 per cent. of ash will be—

$$P = 7257\ (1 - 0{\cdot}48 - 0{\cdot}1) - 1114 \times 0{\cdot}48.$$
$$= 2500 \text{ B.T.U. nearly.}$$

Allowing an efficiency of 40 per cent. we shall have $\frac{2500 \times 0{\cdot}4}{1140}$ i.e. 0·88 lbs. of steam per pound of combustible consumed.

Actual trials have given 0·82 pound of steam for every pound of tan bark, and 0·90 pound of steam for every pound of sawdust consumed.

These combustibles burn much better when they are mixed with oil. It is difficult to utilize them properly, as part of the burning material is carried into the smoke-box.

Peat and Turf.

Turf (mossy, fibrous, or brittle) results from the

decomposition of vegetable matter. This is easily seen in the upper layers, but a little way down in the denser and darker lower layers the remains of the vegetation can no longer be noticed.

Turf dried by being placed in the air contains about 20 to 30 per cent. of water, depending on the locality, and from 10 to 25 per cent. of ash. The weight of a cubic foot varies from 40 to 48 pounds. The combustible is light and spongy, obstructing the free passage of the air. It burns badly, and its smoke has a pungent and disagreeable odour. After grinding and washing it is made into briquettes.

Turf improves by being dried at 212° F.; above this temperature it decomposes. It has, however, to be used immediately after being heated, as it re-absorbs moisture very rapidly.

According to Regnault and Marsilly, dry turf contains on an average 0·57 of carbon, 0·06 of hydrogen, and 0·37 of oxygen and nitrogen.

The free hydrogen is therefore $0{\cdot}06 - \frac{0{\cdot}37}{8} = 0{\cdot}0137$.

The calorific power will be given by

$$P = 0{\cdot}57 \times 14500 + 0{\cdot}0137 \times 62000.$$
$$= 8114.$$

And the true calorific power will be

$$P' = 8114 - 1114\ (9 \times 0{\cdot}06).$$
$$= 7512.$$

If the turf contain 8 per cent. of ash, and 25 per cent. of water,

$$P' = 7512 \times 0{\cdot}67 - 1114 \times 0{\cdot}25.$$
$$= 4755.$$

Carbon from Turf.

This is obtained by carbonizing layers of turf in tubs or ovens made of stone ware or of sheet iron. The return is from 40 to 45 per cent. of a carbon containing from 15 to 20 per cent. of ash. The gases that come from the combustion of the carbon retain the characteristic odour of burning turf. The carbon from Essones has 18 per cent. of ash, and a calorific power equal to $0{\cdot}82 \times 14500 = 11890$.

Lignites.

These combustibles mark the transition stage between peat and coal. They are sometimes brown with a woody texture, and have an earthy appearance, sometimes black with a woody texture, and sometimes homogeneous with a resinous fracture. These last are similar to coal. Regnault distinguishes between the imperfect lignites or the fossil woods and the perfect lignites or woods passing into bitumen. They are characterized by the proportion of oxygen, hydrogen, and carbon (or coke) which they contain.

The calorific power P of the pure combustible has been determined directly by Scheurer-Kestner and Meunier.

Nature.	Coke %	$\frac{O}{H}$	H	P	P_1
Imperfect lignite.	75	5 to 6	4	11,520	9,000
Perfect lignite.	65 to 70	4	4	12,780	9,900
Bituminous.	35 to 40	1 to 2	6	14,040	10,800

With 0·08 of water and 0·10 of ash for the three

varieties, we obtain the true calorific values P'. Also for perfect lignite we have—

$$P' = 12780 \times 0{\cdot}82 - 1114\ (0{\cdot}08 + 9 \times 0{\cdot}04).$$
$$= 10480 - 490.$$
$$= 10000 \text{ B.T.U. nearly.}$$

COALS.

There are a great number of varieties.

Regnault and Gruner divided them into five classes :

1. Non-caking coal (long flame).
2. Gas coal.
3. Coking coal.
4. Coking coal (short flame).
5. Anthracite (short flame).

Non-caking Coal.

Scotch coal and Sandkohle (Germany) are of this kind. They give 60 per cent. of a coke which is pulverulent or only slightly adherent, and they burn with a long flame which lasts only a short time. These coals are put straight on the bars, but they give less heat than the others we mention below. The mass of a cubic foot averages 44 pounds. They are rarely used in France; the French coals which resemble them most closely are those of Saint Éloi, those of the upper strata at Blanzy, and those of Montceau.

Gas Coal. Cherry Coal.

These coals form a cake on the fire without choking it. They are the best coals for steam-raising purposes and for making gas. The *flenu* of Mons and the Cannel coal of Lancashire are the best qualities. They are more abundant in France than the pre-

ceding, and form the upper beds of the Pas-de-Calais, of the Loire, of Commentry and Blanzy.

Coking Coal. Backkohle (Germany).

This coal is of a lustrous black colour; it is not very hard, and splits in layers. In the fire it cakes and gives a light vapoury smoke. It is not suitable for metallurgical operations. A good coke is made from this coal. On the grate it obstructs the free circulation of the air and burns the fire bars, but it gives out a great deal of heat. This coal is plentiful in France, in the basins of Saint-Étienne, of the Nord, and of the Pas-de-Calais. A cubic foot weighs between 44 and 50 pounds.

Coking Coals (short flame).

These coals block the fire less than the preceding, and give a harder coke. They are very suitable for metallurgy. In Belgium they are called hard (dures) because they last (durent) a long time in the fire. They are pulverulent.

In France they are found at Creusot, Saint-Étienne, Brassac, Huy, le Gard and le Nord.

M. Delautel has made at Brest comparative tests of various kinds of this coal.

Taking the calorific power of Cardiff coal as unity, he has found—

Coal d'Anzin . .	1·05 to 1·01.
„ Roche-la-Molière	0·95 to 0·94.
„ Ordinary la Loire . .	0·90.
„ Newcastle . . .	0·84.
„ Blanzy (Montceau). .	0·78.
„ Long flame (Montceau) .	0·74.

Anthracite (short flame).

Anthracite is of a dull, black colour, is difficult to ignite, and burns slowly without smoke. When rapidly heated it is apt to break into pieces. The coke got from it is pulverulent. It is rarely used in France, and is not burned on fire bars. A cubic foot of it weighs 53 lbs.

The *calorific power* of various kinds of coal has been measured directly. The following table B gives the mean value of (1) the elementary composition, and (2) the return in coke of the five types of coal which we suppose pure. It also gives (3) the calorific power P', and (4) the quantity of water taken in at 32° F., which would be converted into steam.

B.—*Mean Results of Five Types of Coal.*

Nature of Coal.	Percentage Elem^y. Comp. C	H	O	Ratio. $\frac{O}{H}$	Mean Cal. Power of Pure Coal.	Water initially at 32° Vaporized at 223·6° per pound of coal.	Coke. %	Nature.
Non-Caking Coal (long flame).	75 to 80	5·5 to 4·5	19·5 to 15	$\frac{4}{3}$	14,400 to 15,300	6·7 to 7·5	50 to 60	Pulverulent or only slightly coherent.
Gas Coal.	80 to 85	5·8 to 5	14·2 to 10	$\frac{3}{2}$	15,300 to 15,660	7·6 to 8·3	60 to 68	Caked and porous.
Coking Coal.	84 to 89	5 to 5·5	11 to 5·5	$\frac{2}{1}$	15,840 to 16,740	8·4 to 9·2	68 to 74	Caked, but with crevices.
Coking Coal (short flame).	88 to 91	5·5 to 4·5	6·5 to 5·5	1	16,740 to 17,280	9·2 to 10	74 to 82	Caked and compact.
Anthracite (short flame).	90 to 93	4·5 to 4	5·5 to 3	1	15,840 to 17,100	9 to 9·5	82 to 90	Brittle or pulverulent.

For a bituminous coal containing 0·02 of water and 0·10 of ash, the proportion of hydrogen will be $0{\cdot}88 \times 0{\cdot}05 = 0{\cdot}044$ and

$$\begin{aligned} P' &= 15570 \times 0{\cdot}88 - 1114\ (0{\cdot}02 + 9 \times 0{\cdot}044) \\ &= 13700 - 1114 \times 0{\cdot}616 \\ &= 13000 \text{ nearly.} \end{aligned}$$

This is how we have calculated P' in Table D.

We make a direct analysis of coal as follows—after drying we find the proportion of water hygrometrically absorbed, we distil the coal in a closed vessel and find the quantity of coke produced. Then burning what is left over after the distillation we find the ash.

The drying is done at a temperature of 230°, about a quarter of an ounce of coal being put in a beaker or between two watch glasses.

The burning is done by placing about one-eighth of an ounce of coal well pulverized in a porcelain capsule in a small furnace suitably warmed to a white heat so as to prevent the formation of coke, which retards the process. We experiment on several capsules at once and take the mean of our results. The burning lasts a quarter of an hour for the coke and two hours for the coal.

The experiments of Scheurer-Kestner and Brix (Table C) agree with the preceding table.

Taking 1152 thermal units for a pound of steam at 233·6° F., we have deduced the efficiency K of each type of pure coal. It will be seen that we can adopt 0·6 as the mean efficiency of coal.

Coal which contains carburetted hydrogen is liable to spontaneous combustion when it is piled in large

heaps, especially if damp or if it contain pyrites. It is necessary in this case to keep it well ventilated.

If it contains too great a percentage of pyrites it may be necessary to keep it under water.

Gas coal alters rapidly in the air; the loss of the gas may amount to as much as 30 per cent. per month when left exposed.

In practice the proportion of ash is from 6 to 10 per cent. with medium-sized lumps, and attains from 10 to 20 per cent. with mixed coal of all sizes, depending on the skill of the stokers and the nature of the coal.

C.—*Results of Vaporization* (*Brick Furnaces*).

Nature of Coal.	Proportion in 100 of Coal.		Water vaporized at 233·6° F. per. lb. of coal.		Mean for Pure Coal.	Efficiency.
	Water.	Ash.	Commercial.	Pure.		
1. *Long Flame Coal.*					$7·3 \times 1152 = 8410$	$\frac{8410}{14800} = 0·57$
Mine Gerhardt (Saarbrück)	5·10	6·84	6·85	7·78		
Mine Leopold (Silèsie sup.)	4·10	5·1	6·1	6·72		
Louisenthal (Saarbrück) .	3·57	12·28	6·06	7·29		
Montceau (Saône-et-Loire)	4·97	10·28	6·2	7·41		
2. *Gas Coal.*					$8 = 9216$	$\frac{9216}{15200} = 0·6$
Friedrichsthal (Saarbrück) .	1	12·7	6·31	7·73		
Altenwald	2·54	13·5	6·95	8·27		
3. *Coking Coal.*					$8·8 = 10140$	$\frac{10140}{16600} = 0·62$
Ronchamp	1·09	16·19	7·62	9·16		
Le Président (Saarbrück) .	1·4	2·28	8·11	8·47		

Briquettes.

The good qualities are moulded under pressure and, at a high temperature, into blocks; the poor qualities are

made into blocks with pitch. Their calorific power is calculated from that of the coal of which they are made. We suppose the coal to be pure and correct for moisture and ash.

Coke (Table B).

Coke is made from coal by subjecting it to a high temperature in a suitable oven so as to get rid of its volatile constituents. The amount of ash it yields varies from 4 to 15 per cent., or even more, as we go from the large lumps to the pulverized coke; and it depends, of course, on the coal from which it was made. It contains from 2 to 10 per cent. of water.

Small coal which has been well washed gives purer and more highly priced coke than that produced without washing the coal.

The cubic foot weighs from 24 to 30 lbs.

Gas coke weighs 18 lbs. per cubic foot.

The calorific power of coke has not been determined directly; we can deduce it from that of carbon by the formula—

$$P = 14500\ (1 - a - b).$$

According to M. de Marsilly dry coke contains on an average 0·04 of hydrogen and 0·06 of ash. For a coke containing 0·02 of water and 0·10 of ash the proportion of hydrogen is reduced to 0·035, from which

$$\begin{aligned} P' &= 14500 \times 0{\cdot}88 - 1092\ (0{\cdot}02 + 9 \times 0{\cdot}035) \\ &= 12760 - 1092 \times {\cdot}335, \\ &= 12400 \text{ nearly.} \end{aligned}$$

Anthracite.

Anthracite is more lustrous than coal; it does not

soil the fingers, and only burns well at a high temperature and when spread in a thick layer.

French anthracites are not abundant; they crumble in the fire and contain on an average 2 per cent. of water and 4 per cent. of ash.

The calorific power of the pure combustible varies from 14,760 to 15,840 B.T.U.

The cubic foot weighs about 30 pounds.

In Table D we have tabulated the number of pounds of steam produced by different combustibles at a pressure of six atmospheres, calculated on the assumption of a 60 per cent. efficiency (except in the case of tan bark). The numbers are given in the last column, and the heat required per pound of steam at this pressure is 1,170 B.T.U.

D.—*Calorific Powers.*

	% Water.	% Ash.	Calorific Power. P.	Calorific Power. P^1.	Volume of gas V. in cubic feet.	Volume at 572° F.	Weight of water in pounds vaporized in practice at 6 atmos.
Wood dried at 284° F.		2		7,290	160	336	3·75
Wood (ordinary) . .	30	2		4,768	133	279	2·5
Tan bark . . .	48	12		2,537	114	239	0·9
Peat	25	8	9,133	5,051	140	294	2·8
Lignite (perfect) . .	8	10	12,770	10,000	176	370	5·12
Coals.							
Non-caking . . .	4	8	14,850	12,860	222	465	6·5
Gas coal	2	10	15,570	13,320	256	536	6·83
Coking coal . . .	,,	,,	16,290	13,860			7·1
Coking coal (short flame)	,,	,,	17,000	14,510			7·34
Half Anthracite (h=4)	,,	,,	16,460	13,860	268	568	7·2
Anthracite . . .	2	4	17,530	15,960	282	590	7·3
Wood Charcoal . .	6	4	12,860	12,860	256	536	6·5
Coke (good quality)	2	5		12,440	273	573	6·4

For long flame coal (non-caking) we have

$$\frac{13320 \times 0{\cdot}6}{1170} = 6{\cdot}83 \text{ lbs.}$$

The weights of water vaporized are those we would get in everyday work with a tubular boiler.

§ 2. Combustion.

The Volume of Air required for Combustion.

Ordinary atmospheric air contains 79 per cent. of nitrogen and 21 per cent. of oxygen.

Oxygen weighs 1·43 oz. per cubic foot, and air weighs 1·29 oz. per cubic foot at the standard temperature (32° F.) and pressure (29·9 inches of mercury).

It follows that one ounce of oxygen is contained in $\frac{1}{0{\cdot}21 \times 1{\cdot}43}$, i.e. 3·33 cubic feet of air at the standard temperature and pressure. For carbonic acid gas containing 72·73 per cent. of oxygen and 27·27 per cent. of carbon, the amount of oxygen necessary to burn one ounce of carbon is $\frac{72{\cdot}73}{27{\cdot}27}$, i.e. 2·667 ounces.

Hence $3{\cdot}33 \times 2{\cdot}667$, i.e. 8·8 cubic feet of air per ounce of carbon, are required.

Since water is composed of 88·9 per cent. of oxygen and 11·1 per cent. of hydrogen, the amount of oxygen necessary to burn one ounce of hydrogen is $\frac{88{\cdot}9}{11{\cdot}1}$, i.e. eight parts of oxygen to one part of hydrogen. Hence we require $8 \times 3{\cdot}33$, i.e. 26·64 cubic feet of air per ounce of hydrogen.

When we know the quantity of carbon C and of free

hydrogen contained in one ounce of the combustible matter, then we can find the volume of air necessary for its combustion by means of the formula—

$$V = C \times 8{\cdot}88 + 26{\cdot}64\left(H - \frac{O}{8}\right).$$

In this formula V is in cubic feet, C, H and O are in ounces.

Example.

Suppose that we have one pound of wood containing 30 per cent. of water, and that its constituents are

Carbon . . .	0·35.
Hydrogen . . .	0·042.
Oxygen and Nitrogen	0·294.
Water . . .	0·3.
Ash	0·014.

The free hydrogen is $0{\cdot}042 - \frac{0{\cdot}294}{8} = 0{\cdot}005$.

The volume of air required to burn a pound of wood will be—

$$0{\cdot}35 \times 8{\cdot}88 \times 16 + 0{\cdot}005 \times 26{\cdot}64 \times 16$$
$$= 51{\cdot}84.$$

For dry coal containing 12 per cent. of ash and water and 88 per cent. of pure combustible material, we get from Table B the average values of the constituents to be as follows:—

$$\left.\begin{array}{lr} \text{Carbon . . } & 0{\cdot}775 \times 0{\cdot}88 = 0{\cdot}682 \\ \text{Hydrogen . . } & 0{\cdot}05 \times 0{\cdot}88 = 0{\cdot}044 \\ \text{Oxygen . . } & 0{\cdot}175 \times 0{\cdot}88 = 0{\cdot}154 \\ \text{Water and Ash} & = 0{\cdot}12 \end{array}\right\} = 1{\cdot}000$$

The free hydrogen will be—

$$0{\cdot}044 - \frac{0{\cdot}154}{8} = 0{\cdot}025,$$

and the volume of air necessary for the combustion of the one pound of coal will be—

$$0{\cdot}682 \times 8{\cdot}88 \times 16 + 0{\cdot}025 \times 26{\cdot}64 \times 16$$
$$= 108 \text{ cubic feet.}$$

In practice two or three times this amount is allowed, as the combustion is never perfect and the door of the furnace is frequently opened.

According to the experiments of Scheurer-Kestner and Meunier the maximum efficiency of combustion occurs when the air is about 33 per cent. in excess of the calculated quantity required.

After the combustion the volume of the gas is the same as that of the air, as carbonic acid gas has the same volume as the oxygen which formed it, but it is increased by the volume of the vapour formed by the hygrometric water in the fuel and by the combustion of the hydrogen.

The volume of this vapour is about 9·6 cubic feet for each pound of peat or wood in the dry state and 12·8 cubic feet when they are moist. For coal it is about 6·4 cubic feet per pound.

Table D gives as the required volume of gas V double the theoretical value plus the volume of the vapour.

For example, we have for dry coal—

$$V = 2 \times 108 + 6{\cdot}4$$
$$= 222{\cdot}4.$$

The volumes of air got from Table D apply to furnaces with a free draught. It is generally accepted that in the case of forced draught, whether by steam blast or fan, the volume of air necessary per pound of fuel is six or seven tenths of V.

The final volume V_t of the gases in the chimney is given by—

$$V_t = V[1 + 0{\cdot}002\ (t-32)]$$

where V is their volume at 32° F. and t is the temperature in the chimney.

For Example—

$$t = 302\ ;\ V_t = 1{\cdot}54\ V.$$
$$t = 402\ ;\ V_t = 1{\cdot}74\ V.$$

Combustion in the Furnaces.

We have seen that carbon and hydrogen are the only elements whose combustion produces heat. The carbon is fixed (coke and wood charcoal) or is contained in volatile hydrocarbons. Coke and wood charcoal can be called the solid fuels, as they only burn on the surface which is made luminous. The fuels which contain hydrocarbons are decomposed by the heat, and combustible gases are disengaged which burn but are only luminous at the surface of contact with air.

On subdividing the fuel and consequently the flames, we increase the surface of contact with air, and the combustion is more rapid and more complete. The heat produced by burning a substance is dissipated (1) by the current of air which takes place round the body and (2) by radiation.

According to Peclet the ratio of the radiating power to the total calorific power is about 25 per cent. for fuels which burn with a flame and about 50 per cent. for carbons.

The following table gives the weight of oxygen and the volume of air which contains that weight necessary

to burn one pound of hydrogen or one pound of carbon, the latter of which is transformed into carbon monoxide $C\,O$ or carbon dioxide $C\,O_2$ and the number of thermal units produced by the combustion:—

Combustible.	Theoretical Requirements.		B.T.U.
	Pounds Oxygen per pound Fuel.	Cubic feet of Air	
Hydrogen gas	8	427	62,032
Carbon perfectly burned ($C\,O_2$)	2·666	142	14,500
Carbon imperfectly burned ($C\,O$)	1·333	71	4,400
Difference			10,100

Combustion of Carbon C.

At the commencement of the combustion one pound of carbon C unites with 2·666 lbs. of oxygen and forms 3·666 pounds of $C\,O_2$, at the same time giving out 14,500 B.T.U. The volume of the $C\,O_2$ is the same as that of the air which formed it, but its density is greater. The combustion is then complete. If, however, there is not sufficient air, the 3·666 lbs. of $C\,O_2$ absorb one pound of carbon and form 4·666 lbs. of $C\,O$. The heat disengaged by the two pounds of carbon transformed into $C\,O$ is

$$2 \times 4400 = 8800.$$

The loss of heat resulting from the absorption of the second pound of carbon is therefore

$$14500 - 8800 = 5700.$$

If now we supply to the 4·666 lbs. of CO the oxygen required to complete the combustion—namely

2·666 lbs., it will burn with a blue flame, and we shall have 7·333 lbs. of CO_2.

The heat set free will be 2 × 10100 = 20200, which, added to the heat produced by the CO, namely 8,800, gives 29,000 units of heat. This is exactly equal to the heat set free by the complete combustion of two pounds of carbon (2 × 14500).

Combustion of Hydrocarbons.

These gases, mixed with a sufficient quantity of air, burn with a blue flame producing carbon dioxide CO_2 and water in the same way as gas for lighting purposes. If, however, they are raised to a red temperature before their mixture with the air required for combustion they are decomposed. The hydrogen burns first, and a part of the carbon is set free. This carbon burns in its turn if there is sufficient oxygen, and if the temperature of the mixture is sufficiently high.

On the other hand, when the combustion is incomplete, and the carbon remains suspended in the gas, then smoke will be formed, which, on cooling, deposits soot. *Once formed, smoke cannot be burned.*

In order to avoid its production it is necessary to mix with the combustible gas a sufficient quantity of air at a sufficiently high temperature, so that the combustion may be complete.

This very simple rule, however, cannot be entirely realized in practice, at least with ordinary furnaces. In a gas-producing plant we can consume nearly all the smoke, but this kind of plant is not suitable for continuous work over lengthened periods.

COMBUSTION

Management of the Fire.

We shall only discuss in this place the combustion of coal. We know that the gas resulting from the combustion of the air flowing in must be at a very high temperature in order to obtain the most perfect combustion. The turning back of the flames upon the fire bridge produces this heating of the gases, and the combustion is better than when the gas rises vertically from the fire-grate. The combustion is completed in the space behind the furnace called the combustion chamber. It is necessary to prevent the flames playing on the tubes of the boiler before the combustion is complete, as otherwise the gases are cooled, and large quantities of smoke and soot are produced. The combustible gases being carried up the chimney often take fire on coming in contact with the open air.

Attempts have been made to increase the heating of the gas and to suppress smoke by injecting steam or air either at the sides of the furnace, or at the fire-bridge, or the door, etc., either continuously or intermittently. We shall not describe all the systems for consuming smoke. None of them up to the present have been completely successful.

An excess of air supply is preferable to an insufficient supply, for the loss of heat due to heated air passing up the shaft is less in the former case than it is in the latter, owing to the imperfect combustion. Smoke consumption needs a large supply of air; it is therefore not the most economical. The combustion cannot be perfect, and hence there is always some smoke formed.

A clever stoker can produce the same results as the best mechanical devices invented. Instructions on this subject are given by the Seine Hygienic Council, and we abstract from them the following:—

"The origin of smoke is in the volatile products given off abundantly from most fuels, such as the various kinds of coal, peat, wood, etc., when subjected suddenly to a high temperature. The most important of these products is carburetted hydrogen, which is itself extremely combustible. In order to burn it, however, we must see (1) that it is supplied with sufficient air, and (2) that the temperature of the mixture is sufficiently high. If these two conditions are not complied with in the furnace or in the flues connected with it, then the carburetted hydrogen undergoes a decomposition, from which there results a large quantity of soot or of carbon in minute particles, which is carried along by the gases passing up the chimney. When we throw on a fire-grate covered with glowing coke a quantity of coal sufficient to cover it to a depth of eight or ten inches, then the particles of coal which are in contact with the coke decompose rapidly; the temperature of the furnace thus suddenly falls and at the same time the passage of air through the grate is obstructed. Neither of the two conditions necessary for the combustion of the carburetted hydrogen is realized, and hence torrents of black smoke issue from the chimney. Under these circumstances the introduction of air by the furnace door, or by any opening above the layer of coal, will have little effect, because the temperature is insufficient to make

the gaseous products burn. The smoke gradually decreases in intensity as the coal gets converted into coke by losing its volatile constituents; the pieces cake together, leaving spaces between them by which the air again gets into the furnace, and so the combustion increases. If we rake the mixture of coal and coke before the distillation of the volatile constituents is complete, we bring portions of coal not yet carbonized into contact with fragments of glowing coke, and so the distillation becomes more rapid, and the chimney begins to smoke again.

"Furnaces which have a grate area sufficiently large to allow the coal to be spread out on it in a thin layer, preferably even not covering it, give little smoke, especially if the coal be put on in small quantities at a time, and if the stoker is careful to put the new coal at the front part of the grate, so that the gaseous products of the distillation arrive at the flues only after they have passed over the coke in the back part of the furnace, where sufficient spaces are left for the entry of the air. The production of smoke is increased when the grate area is too small for the quantity of coal which has to be burned on it in a given time, and when the stokers manage it badly, putting large quantities on at long intervals. Other things being equal, the smoke will be more abundant the more bituminous the coal, and the more it cakes. The dry coals from some parts of the North of France, and from the neighbourhood of Charleroi in Belgium, give very little smoke in ordinary furnaces when burned with ordinary care. Coke produces no smoke; the gases which flow from the top of the chimneys of furnaces

supplied with this material are colourless, carrying with them only a little extremely fine dust."

Coal containing little carburetted hydrogen—as for example, Cardiff coal—gives very little smoke, and has the great advantage of not being liable to spontaneous combustion even when accumulated in great quantities.

When we throw a small piece of coal about the size of a finger on the furnace, it opens out gently as it burns, the middle remaining unchanged until the combustion begins to act there. It is not necessary, therefore, to rake the furnace fire often, since the unburnt portions of coal can drop between the fire bars into the ash pit.

The thickness of the layer of coal on the grate must always be uniform, and be proportional to the draught, but it should not be less than 4 or greater than 6 inches. The stoker should regulate by the flame, which is white when the combustion is good. A red flame is a sign of incomplete combustion due to the layer of fuel being put on too thickly. We feed the fire by putting on the fuel where the layer appears too thin. Large pieces of coal must be broken by the pick in the direction of the cleavage, so as to avoid the formation of dust. The dust can be thrown in a thin uniform layer on the fire when it is in full blaze. We must take care not to put too thick a layer at one point, nor to put it where the fire is not very active.

With very large grates we obtain the best combustion by stoking each half of the grate at alternate intervals,

§ 3.—Steam Trials.

The commercial efficiency of the engine and boiler depends not only on how the steam is utilised in the engine, but also on the steaming power of the boiler.

If we measure the weight of the water injected into the boiler and the weight of fuel burnt, at the same time that we measure the indicated or brake horse power, we shall obtain complete data to determine the actual efficiency. We also obtain the steaming power of the boiler; that is, the weight of steam produced per pound of fuel. The quantity of steam produced per pound of fuel depends on the quality of the fuel, the design of the apparatus, the way it is treated, and especially on the smartness of the stoker.

It has to be remembered also that the results of steam trials are only comparable when they have been made in the same furnace for different fuels or in different furnaces for the same fuel, and in the two cases by the same stoker. When the nature of the fuel that is to be used in the test is specified the contractor determines the percentage efficiency that he can guarantee, knowing the calorific power of the fuel, and he designs his furnace so that the combustion may be as perfect as possible.

When the nature of the fuel is not specified he will naturally employ that which will produce the greatest heat for a given weight.

From the point of view of commercial production it is obvious that the best fuel is that which produces the greatest quantity of steam for a given price. Suppose, for example, that we have to test two kinds

of coal in the same furnace and under the same conditions.

Suppose that a thousand pounds of one fuel costs the user ten shillings, and turns 6,500 lbs. of water into steam. We obtain $\frac{6500}{10} = 650$ lbs. of steam for one shilling. Suppose also that a thousand pounds of the other fuel cost eight shillings and produce 5,000 lbs. of steam; then we get $\frac{5000}{8} = 625$ lbs. of steam for one shilling.

Hence the dearer coal is the more economical. If the steam produced works an engine of 100 horse power, taking 25 lbs. of steam per horse power hour and working ten hours a day for 300 days in the year, then the cost of the fuel per annum will be—

With coal at ten shillings per 1,000 lbs.,

$$\frac{300 \times 10 \times 100 \times 25}{650} = £576\ \ 18s.$$

With coal at eight shillings per 1,000 lbs.,

$$\frac{300 \times 10 \times 100 \times 25}{625} = £600.$$

The annual gain will therefore be about £23. Also we shall need less storage space. It will be seen at once that these calculations have a direct bearing on practice.

Trials of this kind have been made by P. Ducos upon English coal. The pressure of steam in the boiler was about five atmospheres absolute, or about 69 lbs. effective pressure, and the feed water was brought back at 90° F.

Coal.	Price in Shillings.			Steam per lb. of Coal.
	Per Ton.	Per 2,200 lbs. of Steam.	Compared with Cardiff.	
Cardiff . . .	30	3·833	1	7·816
Liverpool . .	26	4·876	1·27	5·332
Newcastle . .	29	5·261	1·37	5·507
Coke with 6 per cent. of water	28	5·259	1·37	5·324

Cardiff, although the dearest coal, is the most economical.

The Management of Steam Trials.

Readings should not be taken until the fire has got well alight, so that the losses due to heating the brickwork, etc., may be the same as in ordinary working. The trials should be over as long an interval of time as possible. They can be made without upsetting the ordinary work of the factory. At the commencement of each test we must thoroughly clean out the grate and the flues, then when the furnace is burning in its normal working condition we must calculate roughly the quantity of coal in the grate, and make allowance for this at the end of the test.

In general we take the total consumption of fuel to be that burned during the test, together with the coal left on the grate at the end of the test, less the quantity of coal on the grate at the beginning of the tests, and the unburnt coal that has dropped through the fire bars. A preliminary test on the fuel is made to determine how much ash and water it contains, from which we can deduce the ratio of the pure coal in a given mass of combustible.

The measurement of the water can be made directly

in any of the following ways—(1) from the number of strokes and the dimensions of the feed pump, (2) by a meter, (3) by automatic feeders, (4) by graduated tanks. This last method is the one employed in very accurate tests. In Fig. 176 the arrangements for measuring the water are shown.

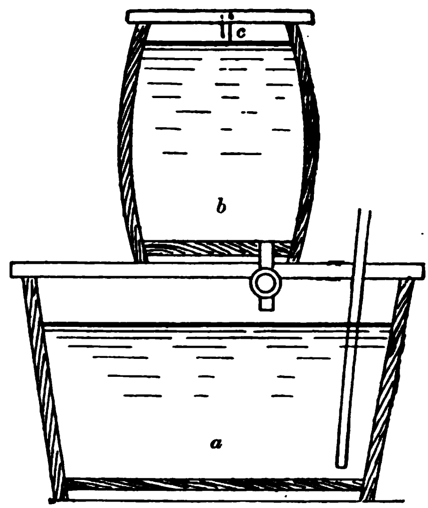


Fig. 176.

The tube of the feed pump is immersed in a closed reservoir *a*. This reservoir is surmounted by a smaller one *b*, such as a cask with the top staved in. A stopcock placed at the bottom of this cask permits water to flow into the reservoir *a*.

The cask *b* is first of all weighed empty. It is then weighed full of water up to the index *c*. The difference between these two weights will give us accurately the weight of water in the cask, and this can be replenished as often as necessary. We measure the quantity of water used during the number of hours the trial lasts. We measure also the weight of

the fuel burnt, and hence we find the number of pounds of water evaporated per pound of fuel.

As we can also find the mean indicated horse power during the run from the indicator diagrams, we can deduce the weight of steam consumed per horse power hour or the weight of fuel consumed per horse power hour.

Condensed Water taken over.

We must arrange at the extremity of the steam pipe, just before it comes to the cylinder, a blow-off cock for the condensed water in the pipe. The weight of that water has to be deducted from the weight of the water injected into the boiler during the test.

As we always mean by steam consumption the weight of the dry steam consumed, it is necessary to deduct further the weight of the water taken over bodily into the cylinder (the priming water) from the weight of water injected into the boiler. We have given two methods of measuring the priming water in Chapter X.

Water from the Cylinder Cocks.

The water obtained from the drainage cocks of the cylinder or the cylinder jacket must not be deducted from the observed consumption.

Certain manufacturers give guarantees of consumption of steam, the consumption being the quantity of water injected into the boiler less the weight of the condensed water in the cylinder jacket. Others take this into account by taking ten per cent. off the observed weight. But the user should not allow these restrictive clauses to be put in the specification.

The efficiency can either be given in pounds of steam produced per pound of coal (gross) or per pound of pure coal, the ashes and the contained water being deducted from the gross weight.

Correction.

We arrange so that at the end of the test the water in the boiler may have the same level and be under the same pressure as at the beginning of the test. If this is not the case a correction has to be made.

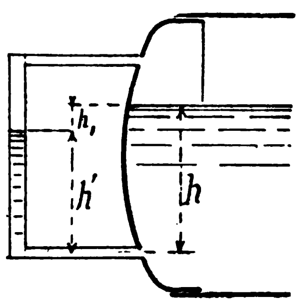


FIG. 177.

We must first of all read the exact levels of the water in the boiler at the beginning and the end of the test. The readings on the tube are not correct, because the water in it is colder than the water in the boiler; it is therefore denser and at a lower level. If we open the cock of the tube for an instant the water in the tube will be renewed and will now have the same temperature as the water in the boiler. Its height is then read before the water has time to cool down.

The difference of the heights $h_,$ (Fig. 177) increases with the difference of the temperatures and becomes important when the water gauge is too far from the boiler.

Suppose that the water in the gauge is at 104° F., then its coefficient of expansion is 0·000259. Let the heights h and h' (Fig. 177) be expressed in feet, then—

$$h_{,} = h - h'$$
$$= h' [1 + 0{\cdot}000259 (t - 104)] - h'$$
$$= 0{\cdot}000259 (t - 104) h'.$$

If $h_{,}$ be expressed in inches, then

$$h_{,} = 0{\cdot}00311 (t - 104) h'$$

Absolute Atmospheric Pressure.	1	2	4	6	8	10
Temperature t.	212	248	291·2	318·2	339·5	356·5
$h_{,}$ in inches.	0·34 h'	0·45 h'	0·57 h'	0·66 h'	0·73 h'	0·79 h'
Density of the Water.	0·955	0·947	0·937	0·930	0·927	0·923

When records have been taken of the exact readings of the water and pressure gauges before and after the trial, then Regnault's tables give us the corresponding temperatures.

To make the required correction we must calculate the number of British thermal units contained in the boiler before and after the trial.

We calculate the volume V that the water would have at 32° F. by the formula—

$$V = \frac{V_t}{1 + 0{\cdot}000259 (t - 32)}$$

The densities given in the table above are found by this formula. We then calculate the number of B.T.U. contained (1) in the corrected weight of the water; and (2) in the metal of the boiler, taking the mean specific heat of iron for temperatures between 0° and 200° to be 0·115 (Dulong and Petit).

We can neglect the different quantities of heat contained in the vapour itself and the effects of expansion.

Knowing the number of units gained or lost by the boiler we can find the number of pounds of steam vaporized, which is the equivalent of this, and so correct our calculation.

Example.

Suppose that an "Elephant" boiler is 3·94 feet in diameter and 49·2 feet long. Suppose also that at the end of the test the level of the water has fallen 3·94 inches, and that the pressure has fallen from 6 to 4 atmospheres. If the feed water is at 59° F., then we shall have—

	Apparent Volume.	Pressure.	Temp. t.	t—59.
Before the Trial.	3,938 gallons.	6	318	259
After the Trial.	3,542 gallons.	4	291	232

Correcting the volumes to 32° F.

Before the trial $3938 \times 0{\cdot}93 = 3{,}663$ gallons at 32°.

After the trial $3542 \times 0{\cdot}937 = 3{,}318$ „ „

Hence the difference $= 345$ gallons.

Before the trial $36630 \times 259 = 9{,}486{,}000$ B.T.U.

After the trial $33180 \times 232 = 7{,}698{,}000$ „

Hence the difference $= 1{,}788{,}000$ „

The weight of the boiler was 33,000 lbs. It has therefore lost

$$33000 \times 0{\cdot}115 \times (318 - 291) \text{ B.T.U.}$$

i.e. 92,460 units. The total loss of heat is therefore 1,880,000. This represents $\frac{1880000}{7200}$, i.e. 250 lbs. of coal at least.

	In Water.	In Coal.
If we have expended during the trial	1,584 gallons.	2,750 lbs.
correction	345 „	250 „
Hence the real expenditure is . . .	1,929 gallons.	3,000 lbs.

Therefore the real evaporative power of a pound of the fuel is $\frac{19290}{3000}$ i.e. 6·4 lbs. of water.

If the level of the water were higher instead of being lower after the test, then the corrections would have to be subtracted from the readings got during the trial.

When a test is made during the everyday running of the boiler, we get too high an evaporative efficiency for the fuel when priming takes place

When the test can be made during an interval in the regular work, we can avoid priming by proceeding as follows:—We let the engine run on light load, but at its normal speed, which is kept as constant as possible, so that the steam flows over slowly but regularly. The surplus steam is lost through the safety valves. We measure the coal and the water as in the preceding test.

We have neglected the fuel burnt during the heating up of the boiler. This error is of small importance during a lengthy test, but we have to take it into account when comparing different kinds of boilers. A boiler which has a great volume of water, like an "Elephant" boiler for example, will take on the first day a large amount of fuel in order to raise steam, but on subsequent days it will take very little, owing

to the large amount of heat stored up in its mass. A boiler which holds little water will, on the contrary, require practically the same amount of fuel each day. The comparison, therefore, will not be accurate unless we take into account the total quantity of fuel burnt per day for several days.

We have already mentioned that the water injected into a boiler can be measured by a water meter. The readings of this meter will give us important information about the working of the boiler, just as the integrating indicator informs us of the working of the engine. For example, if we compare the readings of the meter with those of the recording indicator, and with the quantity of coal burnt, we can see the rate at which the boiler can produce steam, and whether there is priming or not. We can also find out whether incrustations are taking place or not, by comparing the readings with those obtained some weeks previously. These records are also a test of the competence of the stoker.

Testing the Combustion.

When the combustion is perfect the only gases in the flues are carbon dioxide CO_2, which has the same volume as that of the oxygen which formed it, and nitrogen. But in practice there is always a certain quantity of carbon monoxide CO, whose volume is double that of the oxygen which formed it, and in addition there is the oxygen in the unburnt air. The Orsat apparatus, which is only a modification of that of Regnault and Schloesing, elaborated by M. Salleron, the manufacturer, allows us to estimate rapidly the

three gases, O, CO, and CO_2, contained in the products of combustion. A knowledge of these enables us by subtraction to find the nitrogen, and consequently to know what is the volume of air corresponding to a cubic foot of the gaseous mixture.

The apparatus is composed of (1) an aspirator G and a burette M, which serves to measure the volume of the gas at the beginning of the experiment, and after its absorption by each reagent; (2) a series of three absorption vessels A, B and C, in which the

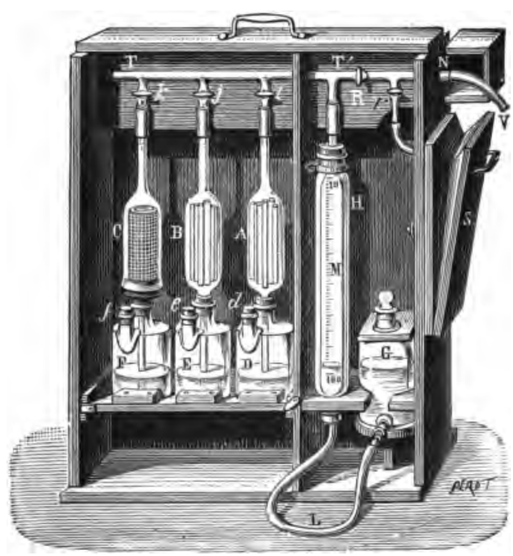


FIG. 178

absorption of each gas is effected by a suitable reagent; and (3) a small bellows S, which can free the tube from stale gases by putting the apparatus in connexion with the flue.

G is a flask containing dilute hydrochloric acid, and hence having no power to absorb carbonic acid gas. This flask is in communication, by means of

a rubber tube L, with the lower part of a graduated vessel M called a burette, which is itself jacketed with a glass vessel full of water, so that the measurements are all made at the same temperature. The upper extremity of the burette is connected to a horizontal glass tube TT', which has a stopcock at R, and is joined to three vertical tubes by cocks i, j and k. These tubes are connected by rubber tubing to the upper extremities of glass vessels A, B and C, called absorption vessels, whilst the lower extremities are immersed in the liquids contained in the flasks D, E and F. All the joints are made air tight by wrapping wire round the rubber tubing, and the openings d, e and f of the flasks are closed with rubber stoppers. The apparatus is connected with the space containing the gas to be analysed by means of the rubber tubing V which is fixed to the horizontal glass tube. Finally a bellows S fixed to the side of the box and communicating with the glass tube by the cock r enables us to extract the air from V.

For our tests we want to know the relative proportions of the carbon dioxide, of the carbon monoxide, and of the oxygen in the products of combustion. The reagents used are (1) a solution of sodium hydroxide which absorbs carbon dioxide; (2) a solution of potassium pyrogallate which absorbs oxygen; and (3) an ammoniacal solution of cuprous chloride which absorbs carbon monoxide.

The absorption vessels A and B contain a large number of glass tubes which are wetted by the solutions and so increase the extent of surface in contact with the gas and thus hasten the absorption.

C contains a roll of copper gauze which, dissolving in the dilute hydrochloric acid, gives rise to cuprous chloride, and thus the cuprous chloride, which is decomposed by the carbon monoxide, is re-formed by the surplus ammonium chloride present, and so the action is continuous.

The potassium pyrogallate and the ammoniacal cuprous chloride absorb oxygen, and hence we must not allow these liquids to come in contact with the air which fills the flasks *E* and *F*. With this object in view the liquids are covered with a layer of petroleum about half an inch thick, for it is absolutely necessary in these operations to prevent any action in the receiving jars.

Suppose now that we have to perform an analysis of the gases in a flue. Opening the cock *R* so that the apparatus is in connexion with the atmosphere, we raise the flask *G*. The acidulated water fills the burette, driving out the air. We then close the cocks *R*, *i* and *j*, open the cock *k*, and take out the stopper *f*. Lowering the flask *G*, the absorption vessel *C* is filled with the solution in *F* owing to the suction produced. We adjust the level of the liquid to the graduated mark on the tube just over the absorption vessel, then we close the cock *k*. We next open the cock *R*, and raising *G*, we again fill the burette; then closing *R*, opening *j* and taking out the stopper *e*, we suck the liquid in the flask *E* into the absorption vessel *B*. Repeating these operations for the third time we fill the absorption vessel *A* with the liquid in *D*.

We now open the cock *R*, then raising the flask *G*, we fill the burette with liquid up to the first graduation

on the upper part of the tube. We then close the cock R and make communication with the flue by the tube N (Fig. 178). We open the cock r and work the bellows sucking the flue gases into the tube V, which is also freed of air or gases formed in the preceding operations. After several seconds, when we are certain that the tube V is filled with the gases, which have to be analysed, we close the cock r and open R, so that the tubes T and V are in communication, and are shut off from the bellows. The water pours into the flask G whilst the burette fills with the gases, and when they have come to the same level we shut the cock R to separate the apparatus from the flue and the bellows, and we make certain that the volume of the gas we are going to experiment on occupies 100 divisions of the burette.

We open the cock i and raise the flask G; the water forces the gas into the absorption vessel A, which contains sodium hydroxide. The bundle of glass tubes multiplying the extent of the surfaces in contact, the carbon dioxide is absorbed; we then lower the flask G, the gas returns into the burette, and the sodium hydroxide again fills the absorption vessel A. We adjust its level by the mark on the tube and close the cock i. We now place the flask G so that the water in it is at the same level as in the burette, so that the gas is at the atmospheric pressure and read the volume the gas occupies. The difference between the readings before and after the operation gives us the volume of the carbon dioxide absorbed by the soda.

We open next the cock j and go through the same operations for the absorption vessel B, which contains

potassium pyrogallate. We thus find the volume of oxygen in the gases.

Finally we proceed in the same manner with the absorption vessel *C*, which contains an ammoniacal solution of cuprous chloride, and this gives the volume of the carbon monoxide. What remains after these three operations is the nitrogen, which cannot be dissolved by any of the preceding reagents.

In order that the absorptions may be complete it is necessary to wash several times the gases in each absorption vessel. We do not pass from one vessel to another until consecutive readings are identical.

The sodium hydroxide used has a specific gravity of 36 deg. Baumé (1·332). The more concentrated the sodium hydroxide is, the more rapid is the absorption of the carbon dioxide. It is necessary then to change the liquid in the flask *D* when the reaction is too feeble; that is, when nearly all the alkali has been transformed into carbonate.

The potassium solution has the same concentration as the sodium. It is advisable to add the pyrogallic acid at the time of the experiment, the quantity being proportional to the oxygen that has to be absorbed. The ammoniacal cuprous chloride is obtained by the solution of the cuprous chloride in a liquid formed of a mixture of two thirds of a saturated solution of chlorhydrate of ammonia, and of one-third of ordinary ammonia (22 deg. Baumé or 1·18).

The cuprous chloride also absorbs oxygen. Hence, in order that the last operation give us the exact quantity of carbon monoxide it is necessary that no oxygen be left unabsorbed by the second operation.

The making of the cuprous chloride being slow, it is sometimes an advantage to replace this by hypochlorous acid ($HClO$) which has also the property of absorbing carbon monoxide. The ordinary commercial crystallised salt is partially altered by exposure to the air. Some of it is put into the flask F and dissolved in hydrochloric acid, and to reduce it to the condition of a protochloride some copper turnings are added to it.

The liquid, which is at first brown, soon loses its colour. We keep it under a layer of petroleum as we have mentioned above.

Suppose that we have initially 100 cubic inches of gas, and let

x = the volume of the nitrogen.

y = the volume of the oxygen and the carbon dioxide.

z = the volume of the carbon monoxide.

We shall have

$$\left.\begin{aligned} x + y + z &= 100 \\ \text{and } \frac{x}{y + \frac{z}{2}} &= \frac{79}{21} \end{aligned}\right\}$$

Eliminating y between these two equations we find

$$100\,x = 7900 - 79\,\frac{z}{2}$$

Hence having found z by analysis, we can determine x.

APPENDIX

Weight of Fuel burnt per Hour.

Boilers with brickwork flues have a mean efficiency $\eta = 0{\cdot}6$ of the calorific power of coal, and for tan-bark and sawdust $\eta = 0{\cdot}4$.

With the same apparatus furnished with reheaters a mean efficiency of $\eta = 0{\cdot}6$ was obtained (Table C).

For fire tube or water-tube boilers—

$$\eta = 0{\cdot}74 \text{ to } 0{\cdot}75.$$

A boiler having both a brickwork furnace and tubes has a mean efficiency $\eta = 0{\cdot}70$.

These ratios have been found taking the calorific power of the pure combustible and using the formula $P' = P(1 - a - b)$.

For fuels which absorb moisture readily, like tan bark or sawdust, we must calculate P' as in table D.

Example.—Suppose that we have to raise per hour 2,200 lbs. of steam at a pressure of six atmospheres (73·5 lbs. gauge pressure) in a fire-tube boiler, the feed water being at 60° F. with coking coal (short flame), $P' = 14{,}510$ (Table D).

The weight of fuel required per hour will be

$$Q = \frac{2200\ (1180 - 60)}{0{\cdot}75 \times 14510}$$

$$= 226 \text{ lbs.}$$

Chimneys and Flues.

Their dimensions are deduced from empirical formulæ.

According to d'Arcet the section S at the summit of a chimney whose height is 33 feet must be one-third of the surface of the grate and correspond to a weight of 66 lbs. of coal burnt per hour per square foot of section.

This rule is in agreement with the more general formula of

APPENDIX

Montgolfier. Let S be the section in square feet, Q the weight of coal burnt per hour in pounds, and H the height of the chimney, then

$$S = 0{\cdot}088 \frac{Q}{\sqrt{H}} \text{ and } S\sqrt{H} = 0{\cdot}088\,Q \quad . \quad . \quad . \quad . \quad (a)$$

H	30	40	50	60	80	100	150	200
$\sqrt{H}$	5·48	6·32	7·07	7·75	8·91	10	12·2	14·1

Other authors suggest the following rules—

$S = \frac{1}{4}$ of the surface of the grate (coal).

$S = \frac{1}{8}$ of the surface of the grate (wood).

Péclet in his *Traité de la Chaleur* has proved—

1. That the draught of a chimney is proportional to $\sqrt{H}$.

2. That there is no advantage gained by letting the products of combustion enter the chimney at a temperature greater than 482° F. It is better to utilize the heat in the boiler and not let the gases escape until they have been cooled to 350 or 400° F.

3. That the real velocity V of the gases in the chimney is less than one-fifth of the theoretical velocity.

For grates burning 21 lbs. of coal per square foot of surface per hour and for $t = 572°$ F., Péclet found that H and V were connected as follows:

H in feet.	32·8	65·6	98·4
V in feet per second	6·0	8·0	9·2

The cubic feet of gas which passed up the chimney per hour in the three cases per square foot of cross section were $6 \times 3{,}600$, i.e. 21,600, 28,800 and 33,120 respectively. Supposing that 290 cubic feet of air were required for every pound of coal consumed, we can burn per square foot in the three cases 74, 96·5 and 107 lbs. per hour respectively.

The ratio of these weights is approximately the same as the ratios of the square roots of the heights ($\sqrt{H}$).

In practice it is generally assumed that

$$S = k\frac{Q}{\sqrt{H}}$$

If the chimney be at some distance from the boiler, k is generally taken as equal to three-fourths of the value given above (0·088).

The table below, giving the heights of the chimneys at the Paris Exhibition of 1878, justifies taking k as 0·088.

If we wish to express S as a function of the horse power N, and if we use m lbs. of coal per horse power hour, the formula can be written—

$$S = K \frac{m\,N}{\sqrt{H}} = K' \frac{N}{\sqrt{H}}$$

If we admit as good practice $m = 5$ lbs. we shall have

$$K' = 0{\cdot}75 \times 0{\cdot}088 \times 5 = 0{\cdot}33$$
$$\text{to } K' = 0{\cdot}088 \times 5 = 0{\cdot}44$$ [1]

Taking the first number we find that the admissible horse-power for a given chimney can be found by the formula

$$N = 3\,S\sqrt{H}.$$

The following table has been calculated from this formula :—

Admissible Power for Given Chimneys.

Diam. in Feet.	Section in Square Feet.	Height of Chimney in feet. H.					
		60	70	80	90	100	150
		Horse Power. N.					
1	0·785	18·3	19·6	21	22·4	23·5	28·7
2	3·14	73	78·7	84·2	89·5	94·2	115
3	7·07	165	178	190	201	212	260
4	12·6	294	315	337	358	378	462
5	19·6	455	491	527	557	589	720
6	28·3	660	712	760	808	850	1040
7	38·5	890	962	1014	1094	1178	1407
8	50·3	1172	1264	1350	1436	1508	1846

[1] The Babcock & Wilcox Co., in their book on "Steam," use K′=0·3, but they increase the diameter of the chimney in order to take account of the friction of the issuing gases. It is, however, not justifiable, as we have shown in this work, to compare this co-efficient with those obtained by Tredgold, d'Arcet, and Péclet, because the modern consumption of coal per actual horse power hour is not comparable to that with which those authors worked.

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For fuels other than coal which require V' cubic feet of air per pound for combustion, coal requiring V cubic feet, we can suppose that the new section S' will be given by the formula

$$S' = S\frac{V'}{V}$$

For peat and wood V' is 128 and for coal V is 256. Hence

$$S' = S\,\frac{8}{16} = \frac{1}{2}\,S$$

The draught being directly proportional both to S and to $\sqrt{H}$, a little consideration will show that it is more economical to increase the section than to increase the height. In determining the section it is necessary to take into account possible future extensions of the plant, but it is a mistake to have it too great, as this would cause down draughts. In towns in France the height is generally fixed at 98·4 feet (30 metres). It must be sufficiently high to prevent the vertical component velocity of the wind having any influence on the velocity of the issuing gases. The horizontal component of the velocity has no appreciable effect on the draught. Other things being the same, draughts are more troublesome with low chimneys than with high ones.

The height must also increase with Q, the quantity of coal burnt per hour, in order that the issuing smoke may be sufficiently diluted with air as not to be a nuisance to the neighbourhood and not spread smuts over those products of the factory which are kept out of doors.

According to an ancient rule which experts still use, a factory chimney must be at least ten feet higher than the roof of any house within a radius of 164 feet (50 metres).

In order to verify the above formulæ we have constructed a table of the heights of the chimneys, etc., at the Paris Exhibition of 1878. The height H is taken from the fire-grate.

The ratios $\frac{q}{S}$ indicate the weight of coal burnt per hour per square foot of the section at the top of the chimney, on the hypothesis that one pound of coal is burnt per square foot of grate surface.

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Chimneys at the Paris Exhibition of 1878.

Boilers at the 1878 Exhibition.	Belleville.	Fives-Lille (Sheet-Iron).	Société Centrale.	Suisse-Belge.	G. Chevalier.	Fontaine.	Boyer-Villette.	Galloway.
	1	2	3	4	5	6	7	8
H., above the grate . .	95	98	108	108	114	120	122	137
D, diameter at top . .	3·9	2·95	2·95	3·28	3·1	2·95	2·85	4·6
S, section in feet . . .	10·5	5·9	5·9	7·3	6·5	5·9	5·6	14·7
g, surface of grate . .	106	41	33·4	105	74	39	37	93
$\frac{g}{S}$ ratio	10	7	5·6	14	11	6·5	6·5	6·5
Coal burnt per square foot	10·2	20·4	20·4	10·2	10·2	20·4	20·4	20·4
$S=0·088\ \frac{Q}{\sqrt{H}}$ gives $D'=$	3·76	3·28	2·62	3·28	3·04	3·1	3·45	4·1

The mean value of $\frac{g}{S}$ for Nos. 1, 4 and 5 is 12. We conclude that those grates burn only half the quantity of coal per square foot of grate surface that the others do. It will be seen that the formula

$$S=0·088\ \frac{Q}{\sqrt{H}}$$

gives values of the diameter D' which are very nearly equal to D, with the exception of the sheet iron chimney (2), where we have had to use the formula

$$S=0·044\ \frac{Q}{\sqrt{H}}$$

At the Sugar Refinery of Bourdon (Puy-de-Dôme) the two principal chimneys are 214 feet high, measured from the level of the grate, and their diameter at the top is 7·4 feet. One of them receives (1) the smoke of six furnaces heating Elephant boilers (Cail) which have 5,200 square feet of heating surface, and (2) the smoke from the furnaces of five tubular boilers (Cail) which have 6,700 square feet of heating surface. Hence there are altogether 11,900 square feet of heating surface. Now the mean value of the coal burnt per hour is 4,180 lbs. and hence 0·35 of a pound of coal is burnt per hour per square foot of heating surface. The formula

$$S = 0{\cdot}088 \frac{Q}{\sqrt{H}}$$

gives a diameter of 5·7. The real diameter 7·4 feet, which is known to be too great, corresponds to the formula

$$S = 0{\cdot}147 \frac{Q}{\sqrt{H}}$$

Chimneys with Forced Draught.

The section in this case is about one-eighth that of the grate surface when the horse power is between two and four, and about one-fifth that of the grate for horse powers between sixteen and twenty. The height varies from 8 to 12 feet for locomotive boilers to 30 feet for boilers which may be described as half-stationary. These heights are, of course, subject to local regulations.

Construction.

The round form is always to be preferred. A little pit just underneath the flues receives the deposit of soot. We employ curved bricks and cement in building the chimney. They are constructed also in stone with Portland cement.

For tall chimneys we place the bricks in layers which have the same diameter for lengths of every three or four feet. It is best to have no cornice, but it is necessary to surround the top with curved metal plates bolted together. The exterior batter should be at least about 3 in 100. Every stage should be 20 to 30 feet in height and every step back about 3·5 inches, the breadth of a brick. With curved bricks these step backs are multiplied, and they may be only about an inch.

The interior diameter diminishes from the bottom to the top of the chimney. In order to diminish the total weight of the construction upon a bad foundation a sheet-iron cylinder is constructed which has a layer of bricks as thin as possible inside.

We place lightning conductors on tall chimneys. The space protected by their influence is a cone whose vertical height is the conductor and the radius of its base equals 1·75 times its height.

Temporary chimneys are often made in sheet iron and painted or, better, galvanized. We make their ends slightly conical to

keep out the rain. They are kept in position by galvanized iron wire-ropes.

Stability of Chimneys.

If we consider a chimney as an elastic solid firmly fixed in its base, we have from the theory of the strength of materials

$$R = \frac{P}{\omega} \pm \frac{v\mu}{I} \quad . \quad . \quad . \quad . \quad (a)$$

Where R = the maximum force per square foot of base.

ω = the area of base $= \pi (r^2 - r'^2)$.

P = the weight of the chimney above base.

$\mu = S\ p \times y$ (see Fig. 179) $= F\ y$.

= the moment of the external forces.

S = the area of the exterior surface $a\ b\ A\ B$.

p = the pressure of the wind per square foot.

I = the moment of inertia of the section.

$= \frac{\pi}{4} (r^4 - r'^4)$ for a round chimney.

$v = \pm r$.

= the distance of the fibres furthest from the neutral axis O.

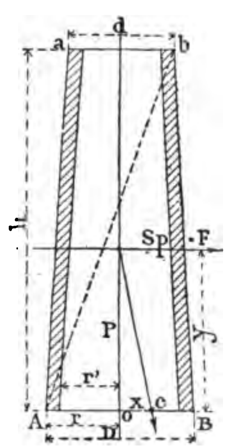

FIG. 179.

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The volume of the chimney shown in Fig. 179 is approximately equal to the mean of the area of its ends multiplied by its perpendicular height. The density of the brickwork is about 106 lbs. per cubic foot.

The pressure exerted by a gale is estimated at 60 lbs. per square foot on a plane surface placed perpendicularly to the direction of the wind in an exposed position. In valleys and when the chimneys are partly protected by other buildings the pressure rarely exceeds 40 lbs. per square foot. It can also be proved that it is only half this amount per square foot of cylindrical surface.

Hence $p = 30$ lbs. or 20 lbs.

Dividing the trapezium $a\ b\ A\ B$ (Fig. 179) into two triangles by joining A and b, we get the following equation to find y, the height of the centre of pressure.

$$Sy = D\frac{h}{2} \times \frac{h}{3} + d\,\frac{h}{2} \times \frac{2h}{3}.$$

$$= \frac{h^2}{6}\ (D + 2\,d).$$

Multiplying by p we get the moment μ—

$$\mu = Sy \times 30 = 5\,h^2\ (D + 2\,d)$$

$$\mu = Sy \times 20 = \frac{10}{3}\,h^2\ (D + 2\ d).$$

Substituting these values of μ in (a) above we get the approximate formulæ—

$$R = \frac{P}{\omega} \pm 6{\cdot}5\ \frac{r\,h^2\ (D + 2\,d).}{r^4 - r'^4} \quad . \quad . \quad . \quad (b)$$

$$R = \frac{P}{\omega} \pm 4{\cdot}5\ \frac{r\,h^2\ (D + 2\ d).}{r^4 - r'^4} \quad . \quad . \quad . \quad (b')$$

For iron chimneys r differs very little from r', and taking $p = 20$ lbs. per square inch we have

$$R = \frac{P}{\omega} \pm 1{\cdot}1\ \frac{h^2 (D + 2\ d).}{r^2 (r - r')} \quad . \quad . \quad . \quad (c)$$

When iron chimneys are steadied by guy ropes, the part of the

chimney below the point of attachment is only subjected to a pressure $\frac{P}{\omega}$.

The force $F = S\,p$ (Fig. 177) produces a compression increasing from O to B which is added on to the compression $\frac{P}{\omega}$.

From O to A, F produces an extension which opposes $\frac{P}{\omega}$ and is a maximum at A. At A there is compression or extension according as R is positive or negative.

Application of Formula to a Chimney 95 *feet high.*

P the weight of the chimney at the foot of the 95 feet is 240,000 lbs.

$$r = 4{\cdot}92 \text{ feet}; \; r' = 3 \text{ feet}; \; d = 4{\cdot}4 \text{ feet};$$
$$\omega = 47{\cdot}3 \text{ square feet and } \frac{P}{\omega} = 5090$$

Substituting these values in (b') we have

$$R = 5090 \;\pm 4{\cdot}5 \frac{4{\cdot}92 \times (95)^2\,(9{\cdot}84 + 2 \times 4{\cdot}4)}{(4{\cdot}92)^4 - 3^4}$$

$$= 5090 \;\pm 4{\cdot}5\,\frac{45000 \times 18{\cdot}64}{505}$$

$$= 5090 \;\pm\; 7540.$$

hence R per square inch

$$= 35{\cdot}3 \;\pm\; 51{\cdot}1.$$

Hence for compression $R = 86{\cdot}4$ lbs. per square inch and for extension $R = 15{\cdot}8$ lbs. per square inch.

Hard bricks well baked, sometimes called " Burgundy " bricks, with hydraulic mortars or the cements which are preferably employed crush under a load of 1,800 to 2,000 lbs. per square inch, and break on extension under a load of 200 to 250 lbs. per square inch. In practice the compression must not exceed 150

to 180 lbs. per square inch, and the extension should not be greater than one-fifth of that for which rupture occurs, i.e. it should not exceed 40 or 50 lbs. per square inch.

Flues.

The section of a flue should be at least equal to that of the chimney. It increases on approaching the furnace because the volume of the gases is greater and because there are usually more abundant deposits of soot. The section for the passage of the gases over the fire bridge is about 60 per cent. of the area of the grate. The first flue under the boiler is generally sufficiently high to allow the smoke and ash carried over to drop into a lower channel. The section of the flues which follow is about 75 per cent. of that of the first or 50 per cent. of the area of the grate.

Furnaces with Ordinary Grates.

Grates are generally raised about two feet from the ground, and are either horizontal or preferably inclined downwards from the front to the back at the rate of 1½ to 2 inches per foot of grate. The bars are fixed loosely on bearers and are generally made of cast iron. Their length varies from 1½ feet to 2½ feet, and they are made thinner towards the lower edge so as to give access to the draught and facilitate the falling of ashes and the cooling of the bar. The space between the bars varies from ¼ to ⅜ of an inch, and the empty space is about a third or a fourth of the total surface of the grate. The thickness of the ends of the bars keeps the dimensions of these spaces fixed, and as a further precaution in the case of long bars a boss is made at their centres to prevent them closing up.

Bars must be free to expand in every direction, and to facilitate this an extremity is often cut on the slant.

Thin bars alter less and subdivide the air better under the fuel. For a given space between the bars (¼ to ⅜ of an inch) they also give more air space for a given area of grate.

The drier the fuel and the more it is divided or the more it divides in the fire like anthracite and dry coal, the more it is necessary to reduce the size of the spaces between the bars.

For large coal these spaces must be wide enough to allow the rake to go through and the bars must be stronger. Grates are

sometimes made of special laminated iron bars. In this case two or three of the bars are riveted together.

The surface of the grate can be calculated by counting that it must burn 8 to 12 lbs. of coal per square foot per hour when the combustion is slow, and 14 to 20 lbs. of coal per square foot per hour for rapid combustion. The latter system is preferable, especially for high powers, as it allows us to use smaller grates, which are easier to manage.

In furnaces which use forced draught, whether steam blast or fan, we can burn from 40 to 80 lbs. of coal per square foot per hour.

In order that a grate may be easy to manage it must not be longer than 5 or 6 feet and broader than about 3 if the combustion be rapid or 5 if it be slow. The wall at the end or fire-bridge rises from 8 inches to 1 foot 4 inches in height above the level of the bars according to the thickness of the layer of fuel.

Single doors are generally 8 inches high and from 8 to 12 inches broad; double doors are from 10 to 12 inches high by from 16 to 24 in breadth. They are mounted on an inner door of sheet-iron and are separated from the grate by a cast-iron base from eight to sixteen inches broad.

The following table indicates the most suitable thickness of the layer on the grate for various fuels, the weight burned per square foot and the shortest distance between the grate and the boiler.

Nature of Fuel.	Thickness of layer (inches).	Pounds burned per square foot.	Minimum dist. from boiler to grate (inches).
Finely divided coal	2 to 3½	0·9 to 1·3	12 to 16
Nuts	4 to 6	1·3 to 2·2	17·5 to 19·5
Dry Coal. . . .	6 to 8	2·2 to 2·85	ditto
Coke	8 to 12	4·4 to 6·6	21·5 to 23·75
Wood	12 to 16	6·6 to 7·7	23·75 to 29·5
Tan Bark. Sawdust .		2·2	ditto
Peat. Turf . . .		6·6	19·5 to 21·5

The distance of the boiler from the grate is a little farther when the surfaces are very large.

When we wish to get more power from a given boiler it is generally more economical to buy a higher grade coal than to

increase the thickness of the layer on the grate, which would make it burn badly and give us a low efficiency.

Grates for sawdust and tan-bark must be large in order to diminish the velocity of the draught, which has a tendency to take the burning particles along with it. Hence we must have large flues and they must be frequently cleaned.

Ash Pit.

The ash pit must offer to the air a free passage at least equal in section to that of the chimney. The use of a water tank underneath the furnace has certain advantages. First, the unburnt pieces of coal can be used over again; then, secondly, the air drawn under the grate is not heated by the burning ashes, the maximum amount of oxygen is furnished for the combustion and the bars are less heated, and finally the stoker can see the state of the fire by watching its reflection on the water. The ash pit must be provided with gates which, when the register is closed, suppress all draught and keep the fire alight from evening to morning.

Boiler.

The thickness of the plates has practically no influence on the transmission of heat from the furnace gases to the water. The heat transmitted is proportional to the difference of temperature between the two sides of the plates. In boilers where the heating is on the exterior surface, the heating from below is much more efficacious than the heating from above. First, because of the convection currents, the hotter water continually rising and the colder water flowing underneath. Hence the difference of temperature between the gas and the water is greater and the bubbles of vapour which are formed rise rapidly, whilst the vapour which accumulates in the upper part as in water-tube boilers circulates slowly and gives up its heat, thus rendering the transmission of heat throughout the mass of the boiler more gentle, as the specific heat of steam is only 0·5. Secondly, because the upper part of a boiler is covered quickly with ashes which prevent the contact with the warm furnace gases. Also it is convenient to make the arches of the furnace rest on the upper side of the boiler.

In the lower surface of the boiler the transmission of the heat

can be modified by incrustations of lime. It appears more active with a thin deposit after the boiler has been in use some time than with new iron plates. The transmission of heat is accelerated by a well designed circulation of the water in the boiler.

The feed water must be injected into the lowest and coldest part of the boiler.

According to theory we ought to utilize the heat as near to the fire as possible, or where the temperature is a maximum, and not allow the products of combustion to escape until they were cooled down as far as practicable. No advantage, however, results from cooling them below 400° F.

The direct heating surface must be as near to the grate as the nature of the fuel permits (see the table given above).

In flues the ratio of the heating surface to that of the brick work must be as great as possible, so that the loss of heat by the brickwork may be a minimum. These losses are small when the furnace is embanked in the ground

Evaporation per Square Foot.

The greater the heating surface for a given quantity of coal burnt, or the less coal burnt for a given heating surface, the higher is the efficiency. But this rule, which is as old as the invention of boilers and has suggested reheaters, has in practice a limit below which the advantage is nothing since the heat absorbed by the furnace is practically constant.

The maximum efficiency appears to correspond to a consumption of 0·35 to 0·4 lb. of coal per square foot of heating surface per hour.

If we burn upon the grate per hour 18 lbs. of good coal per square foot of surface, the heating surface will be about fifty times that of the grate.

If we burn 0·4 lb. of good coal per square foot of heating surface, then the water raised in temperature from 60° F. to 310° F. (6 atmospheres) and evaporated will be

$$0{\cdot}4 \times \frac{14500}{1152} = 5 \text{ pounds per square foot.}$$

Adopting the efficiencies mentioned before, tubular boilers will evaporate

$5 \times 0{\cdot}5 = 2{\cdot}5$ pounds per square foot.

Tubular boilers with reheaters will evaporate

$5 \times 0{\cdot}6 = 3$ pounds per square foot.

"Mixed boilers" will evaporate

$5 \times 0{\cdot}7 = 3{\cdot}5$ pounds per square foot.

Fire-tube boilers will evaporate

$5 \times 0{\cdot}75 = 3{\cdot}75$ pounds per square foot.

These numbers are the ones generally used, and their ratios are 12, 15, 18 and 20. If we increase the evaporation too much there will be priming.

Proportions of Various Types.

The proportions or ratios between the heating surface and the grate area $\frac{C}{S}$ for various types of boiler are given in the table below. Also the ratio between the total heating surface and the volume of water $\frac{T}{O}$ and the ratio of $\frac{S}{s}$; i.e. the ratio of the grate area to the section of the tubes.

The boilers are of five types—

1. Galloway boilers.
2. Half-tubular.
3. Fixed furnace and tubes.
4. Locomotive.
5. Water-tube, heated from the outside.

Volume of Water (O). Heating Surface per Cubic Foot of Water.

In non-tubular boilers the volume of the water may be six or ten times as much as that evaporated per hour.

Suppose that we have an evaporation of 0·065 of a cubic foot of water per square foot of heating surface per hour. The heating surface per cubic foot of water will be

With the ratio 6; . . . . $\frac{1}{6 \times 0{\cdot}065} = \frac{1}{0{\cdot}39} = 2{\cdot}56$

With the ratio 10; . . . . $\frac{1}{10 \times 0{\cdot}065} = \frac{1}{0{\cdot}65} = 1{\cdot}54$

For the half-tubular and the fire-tube boilers this ratio rises

from 3·0 to 4·5. Finally, for water-tube boilers (tubes outside) this heating surface per cubic foot of water rises to 9 and 21.

The small volumes of water reduce the space occupied by the boiler and allow powerful generators to be used in central districts.

Volume of Vapour (V).

This volume is generally from two to two and a half times that evaporated per hour, but as before there is no fixed rule. It is one-fifth the volume of the water in Elephant boilers, although it is often equal to the volume of the water in boilers which produce steam rapidly.

		Heating Surface.				Volumes.					Tubes.	
	Area of Grate.	Boiler and Tubes.	Reheater.	Total.	Ratio.	Water cubic ft.	Steam cubic ft.	Heating Surface per cubic ft.	Water Surface.	Heating Surface per sq. ft. Water Surface.	Section.	Ratio to grate.
	S.	C.		T.	$\frac{C}{S}$	O.	V.	$\frac{T}{O}$	n.	$\frac{C}{n}$	s.	$\frac{S}{s}$
Villette	21	1085	450	1535	53	740	141	2·14	102·1	10·7	—	—
Eschger	13	495	—	495	40	268	98	1·83	46·4	10·7	—	—
Galloway	35·5	1150	—	1150	33	812	230	1·43	172	6·7	—	—
Chevalier G. et. D.	29	850	645	1495	30	635	159	2·44	176	5	—	—
Ditto	28	772	322	1094	29	708	254	1·58	172	4·5	—	—
Half Tubular. Meunier	27·4	1130	—	1130	41	338	152	3·35	79	14	1·3	7·7
Half Tubular. Fontaine	22·5	1180	258	1438	52	460	120	3·05	73	16	1·38	6
Half Tubular. Lebrun	36	1870	64·5	1935	52	530	98	3·66	65	29	2·65	5
Fixed Funnel. Ste. Cle.	14	720	—	720	52	223	77	3·05	48·5	15	0·83	5·6
Fixed Funnel. Farcot	30·5	1720	171	1891	56	495	240	3·8	126	13·6	1·62	6·8
Locomotive. Sulzer	14	450	258	708	33	194	84	3·66	17·7	25	0·95	5·4
Locomotive. Séraphin	22·8	1150	225	1375	50	296	120	4·56	28·7	40	1·3	6·5
Locomotive. Fives Lille	23·6	1240	—	1240	52	148	148	8·25	98	12	1·34	6·5
Water Tube. Barbe Petry	45	1525	—	1525	34	353	152	4·26	130	12	—	—
Water Tube. Mac Nicol	19·4	645	—	645	33	106	95	6·1	30.2	22	—	—
Water Tube. de Naeyer	37·5	1700	—	1700	45	177	106	9·75	—	—	—	—
Water Tube. Belleville	41	1245	—	1245	30·5	53	—	23·5	—	—	—	—

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Heating Surface per Square Foot of the Surface of the Water.

This ratio varies from 5 to 8 and has a mean value of 6·5. The higher this ratio the more violent is the ebullition and the greater is the priming.

Safety Valves.

The diameter of safety valves in inches is determined by the following formula—

$$d=\sqrt{\frac{1\cdot42\,C}{p+8\cdot5}} \qquad \ldots\ldots \qquad (a)$$

where p = the effective pressure in pounds per square inch and C = the total heating surface in square feet.

The breadth of the seat of the valve does not exceed 24 mils. for small boilers and 80 mils. for large boilers.

The diameters of the safety valves in the following table are calculated by means of formula (a) above :—

Heating Surface in square feet.	Diameters of Safety Valves in Inches. Gauge Pressure in lbs. per square inch.								
C.	40	50	60	70	80	90	100	110	120
20	0·77	0·70	0·64	0·60	0·57	0·54	0·51	0·49	0·47
40	1·08	0·99	0·91	0·85	0·80	0·76	0·72	0·69	0·66
60	1·33	1·21	1·12	1·04	0·98	0·93	0·89	0·85	0·81
80	1·53	1·39	1·29	1·20	1·13	1·07	1·02	0·98	0·94
100	1·72	1·56	1·44	1·34	1·27	1·20	1·15	1·09	1·05
120	1·88	1·71	1·58	1·47	1·39	1·32	1·25	1·20	1·15
140	2·03	1·85	1·71	1·59	1·50	1·42	1·36	1·30	1·25
160	2·16	1·98	1·82	1·70	1·60	1·52	1·44	1·38	1·32
180	2·30	2·09	1·93	1·80	1·70	1·61	1·54	1·47	1·41
200	2·42	2·21	2·04	1·90	1·79	1·70	1·62	1·55	1·49
250	2·71	2·46	2·28	2·13	2·00	1·95	1·81	1·73	1·66
300	2·97	2·70	2·50	2·33	2·19	2·08	1·98	1·89	1·82
400	3·44	3·12	2·88	2·68	2·54	2·40	2·30	2·18	2·10
500	3·83	3·48	3·22	3·01	2·83	2·69	2·56	2·45	2·35
600	4·20	3·82	3·53	3·29	3·10	2·94	2·81	2·68	2·58
700	4·54	4·13	3·81	3·56	3·35	3·18	3·03	2·89	2·78
800	4·84	4·42	4·08	3·80	3·58	3·40	3·24	3·10	2·98
900	5·16	4·68	4·32	4·02	3·81	3·60	3·45	3·27	3·15
1000	5·41	4·93	4·55	4·25	4·00	3·80	3·62	3·46	3·32
1250	6·06	5·51	5·09	4·76	4·48	4·25	4·05	3·88	3·72
1500	6·63	6·03	5·58	5·21	4·91	4·66	4·43	4·24	4·07
2000	7·67	6·97	6·44	6·01	5·66	5·37	5·12	4·89	4·70

APPENDIX

COMPARATIVE TABLE OF PRESSURES.

Atmospheres Absolute.	Effective lbs. per sq. in.	Effective lbs. per sq. in.	Atmospheres Absolute.
1	0·000	10	1·68
1·5	7·35	20	2·36
2	14·69	30	3·04
2·5	22·04	40	3·72
3	29·38	50	4·40
3·5	36·73	60	5·08
4	44·07	70	5·77
4·5	51·42	80	6·45
5	58·76	90	7·13
5·5	66·11	100	7·81
6	73·45	110	8·49
6·5	80·80	120	9·17
7	88·14	130	9·85
7·5	95·49	140	10·53
8	102·83	150	11·21
8·5	110·18	160	11·89
9	117·52	170	12·57
9·5	124·87	180	13·25
10	132·21	190	13·93
10·5	139·56	200	14·61

EFFECTIVE PRESSURE AND TEMPERATURE.

Effective Pressure lbs. per sq. inch.	T. Temperature.	Effective Pressure lbs. per sq. inch.	T. Temperature.
0	212° F.	75	322·1° F.
5	227·6 „	80	326·0 „
10	240·3 „	85	329·6 „
15	250·9 „	90	332·8 „
20	260·2 „	95	336·6 „
25	268·3 „	100	339·8 „
30	275·7 „	110	346·0 „
35	282·4 „	120	351·8 „
40	287·5 „	130	357·2 „
45	294·3 „	140	362·4 „
50	299·6 „	150	367·1 „
55	304·7 „	160	371·9 „
60	309·4 „	170	376·2 „
65	313·9 „	180	380·4 „
70	318·1 „	190	384·5 „
		200	388·4 „

APPENDIX

PROPERTIES OF SATURATED STEAM.

Absolute Pressure.		Temperature in degrees F.	Total Heat in lbs. from 32° F.	Weight of 1 cubic ft. in lbs.	V. Volume of 1 lb. in cubic ft.	935·4. / V.
Atmospheres.	lbs. per sq. inch.					
·1	1·469	115·16	1117·06	·004279	233·7	4·002
·2	2·938	140·81	1124·89	·008256	121·2	7·723
·3	4·407	157·10	1128·04	·01157	86·46	10·81
·4	5·876	169·25	1131·77	·01618	61·79	15·14
·5	7·345	179·10	1134·75	·01949	51·31	18·23
·6	8·814	187·30	1139·07	·02335	42·83	21·84
·7	10·283	194·50	1140·52	·02709	36·92	25·34
·8	11·752	201	1143·23	·03058	32·71	28·63
·9	13·221	206·10	1144·99	·03418	29·26	31·97
1·0	14·690	212	1146·60	·03773	26·51	35·30
·1	16·159	216·8	1148·27	·04123	24·25	38·57
·2	17·628	221·3	1149·44	·04477	22·34	41·88
·3	19·097	225·5	1150·72	·04826	20·72	45·14
·4	20·566	229·4	1151·91	·05175	19·32	48·41
·5	22·035	233·1	1153·04	·05524	17·94	52·14
·6	23·504	235·8	1154·12	·05872	17·03	54·93
·7	24·973	239·9	1155·13	·06215	16·09	58·11
·8	26·442	243·1	1155·99	·06560	15·25	62·76
·9	27·911	246	1157·02	·06896	14·50	64·52
2·0	29·380	249·1	1157·90	·07243	13·80	67·75
·1	30·849	251·8	1158·77	·07577	13·20	70·87
·2	32·318	254·5	1159·58	·07921	12·63	74·09
·3	33·787	257	1160·37	·08256	12·12	77·23
·4	35·256	259·7	1161·13	·08594	11·63	79·89
·5	36·725	262	1161·86	·08930	11·20	83·52
·6	38·194	264·4	1162·58	·09268	10·79	86·70
·7	39·663	266·7	1163·27	·09603	10·41	89·82
·8	41·132	268·8	1164·33	·09936	10·07	92·94
·9	42·601	271	1164·58	·1041	9·605	97·38
3·0	44·070	273	1165·21	·1060	9·434	99·15
·1	45·539	275	1165·81	·1092	9·152	102·2
·2	47·008	276·8	1166·43	·1126	8·882	105·3
·3	48·477	278·9	1167·01	·1159	8·630	108·4
·4	49·946	280·7	1167·59	·1192	8·389	111·5
·5	51·415	282·5	1168·14	·1223	8·181	114·4
·6	52·884	284·3	1168·69	·1258	7·949	117·7
·7	54·353	286·2	1169·23	·1291	7·747	120·8
·8	55·822	287·9	1169·75	·1323	7·558	123·8
·9	57·291	289·6	1170·25	·1356	7·372	126·9
4·0	58·760	291·2	1170·75	·1389	7·199	129·9
·1	60·229	292·8	1171·24	·1421	7·037	132·9
·2	61·698	294·3	1171·73	·1454	6·877	136·0
·3	63·167	295·9	1172·19	·1487	6·716	139·1
·4	64·636	297·5	1172·66	·1519	6·585	142·1
·5	66·105	298·9	1173·11	·1552	6·446	145·1

PROPERTIES OF SATURATED STEAM.—*continued.*

Absolute Pressure. Atmospheres.	Absolute Pressure. lbs. per sq. inch.	Temperature in degrees F.	Total Heat in 1 lb. from 32° F.	Weight of 1 cub. ft. in lbs.	V. Volume of 1 lb. in cubic ft.	935·4 / V.
·6	67·574	300·4	1173·56	·1583	6·316	148·1
·7	69·043	301·8	1173·99	·1616	6·187	151·2
·8	70·513	303·7	1174·43	·1647	6·074	154·1
·9	71·981	304·7	1174·86	·1681	5·951	157·1
5·0	73·450	305·9	1175·27	·1712	5·839	160·2
·1	74·919	307·4	1175·69	·1745	5·732	163·2
·2	76·388	308·7	1176·08	·1776	5·630	166·2
·3	77·857	309·9	1176·48	·1808	5·530	169·1
·4	79·326	311·1	1176·84	·1847	5·414	172·8
·5	80·795	312·5	1177·27	·1873	5·340	175·2
·6	82·264	313·9	1177·63	·1904	5·252	178·1
·7	83·733	314·9	1178·01	·1936	5·165	181·1
·8	85·202	316·2	1178·39	·1968	5·081	184·1
·9	86·671	317·5	1178·75	·2000	4·999	187·1
6·0	88·140	318·6	1179·11	·2032	4·921	190·1
·1	89·609	319·6	1179·47	·2064	4·846	193·1
·2	91·078	320·9	1179·81	·2095	4·773	196·0
·3	92·547	322	1180·17	·2127	4·700	199·0
·4	94·016	323·3	1180·51	·2159	4·631	201·9
8·5	95·485	324·1	1180·83	·2191	4·565	204·8
·6	96·954	325·2	1180·97	·2222	4·501	207·8
·7	98·423	328·3	1181·50	·2254	4·437	210·8
·8	99·892	329·5	1181·83	·2286	4·375	213·8
·9	101·361	330·5	1182·15	·2316	4·317	216·7
7·00	102·830	331·5	1182·47	·2347	4·260	220·1
·25	106·503	332·1	1183·26	·2427	4·121	227·0
·50	110·176	334·7	1184·02	·2505	3·992	234·3
·75	113·849	337·1	1184·76	·2584	3·870	241·7
8·00	117·522	339·5	1185·48	·2662	3·758	249·0
·25	121·195	341·7	1186·18	·2740	3·650	256·4
·50	124·868	343·9	1186·86	·2817	3·550	263·5
·75	128·541	347·9	1186·93	·2895	3·453	270·8
9·00	132·214	348·3	1188·19	·2973	3·363	278·1
·25	135·887	350·4	1188·85	·3050	3·278	285·3
·50	139·560	352·4	1189·47	·3126	3·199	292·4
·75	143·233	354·5	1190·09	·3205	3·120	299·7
10·00	146·906	356·5	1190·70	·3282	3·047	307·0
·25	150·579	358·3	1191·27	·3359	2·977	314·2
·50	154·252	360·3	1191·85	·3436	2·911	321·4
·75	157·925	362·1	1192·23	·3513	2·841	329·3
11·00	161·598	364·1	1192·98	·3589	2·786	335·7

APPENDIX

DIAMETERS AND AREAS FROM 1 TO 1000.

No.	0.	0·2.	0·4.	0·6.	0·8.
1	0·7854	1·131	1·539	2·011	2·545
2	3·1416	3·801	4·524	5·309	6·1575
3	7·0686	8·042	9·079	10·179	11·341
4	12·566	13·854	15·205	16·619	18·096
5	19·635	21·237	22·902	24·630	26·421
6	28·274	30·191	32·170	34·212	36·317
7	38·485	40·715	43·008	45·365	47·784
8	50·265	52·810	55·418	58·088	60·821
9	63·917	66·476	69·398	72·382	75·430
10	78·540	81·713	84·949	88·247	91·609
11	95·033	98·520	102·070	105·683	109·359
12	113·098	116·899	120·763	124·690	126·680
13	132·733	136·848	141·026	145·267	149·572
14	153·938	158·368	162·861	167·115	172·034
15	176·715	181·459	186·265	191·13	196·07
16	201·06	206·12	211·24	216·42	221·67
17	226·98	232·35	237·79	243·28	248·85
18	254·47	260·16	265·90	271·72	277·59
19	283·53	289·53	295·59	301·72	307·91
20	314·16	320·47	326·85	333·29	339·79
21	346·36	352·99	359·68	366·44	373·25
22	380·13	387·08	394·08	401·15	408·28
23	415·48	422·73	430·05	437·44	444·88
24	452·39	459·96	467·59	475·29	483·05
25	490·87	498·76	506·71	514·72	522·79
26	530·93	539·13	547·39	555·72	564·10
27	572·56	581·07	589·65	598·28	606·99
28	615·75	624·58	633·47	642·42	651·44
29	660·52	669·66	678·87	688·13	697·46
30	706·86	716·31	725·83	735·42	745·06
31	754·77	764·54	774·37	784·27	794·23
32	804·25	814·33	824·48	834·69	844·96
33	855·30	865·70	876·16	886·68	897·27
34	907·92	918·63	929·41	940·25	951·15
35	962·11	973·14	984·23	995·38	1,006·60
36	1,017·87	1,029·21	1,040·60	1,052·09	1,063·62
37	1,075·21	1,086·87	1,098·58	1,110·4	1,122·2
38	1,134·1	1,146·1	1,158·1	1,170·2	1,182·4
39	1,194·6	1,206·9	1,219·2	1,231·6	1,244·1
40	1,256·6	1,269·2	1,281·9	1,294·6	1,307·4
41	1,320·3	1,333·2	1,346·1	1,359·2	1,372·3
42	1,385·4	1,398·7	1,412·0	1,425·3	1,438·7
43	1,452·2	1,465·7	1,479·3	1,493·0	1,506·7
44	1,520·5	1,534·4	1,548·3	1,562·3	1,576·3
45	1,590·4	1,604·6	1,618·8	1,633·1	1,647·5
46	1,661·9	1,676·4	1,690·9	1,705·5	1,720·2
47	1,734·9	1,749·7	1,764·6	1,779·5	1,794·5
48	1,809·6	1,824·7	1,839·8	1,855·1	1,870·4
49	1,885·7	1,901·2	1,916·7	1,932·2	1,947·8
50	1,963·5	1,979·2	1,995·0	2,010·9	2,026·8

APPENDIX

DIAMETERS AND AREAS FROM 1 TO 1000 (*continued*).

No.	0.	0·2.	0·4.	0·6.	0·8.
51	2,042·8	2,058·9	2,075·0	2,091·2	2,107·4
52	2,123·7	2,140·1	2,156·5	2,173·0	2,189·6
53	2,206·2	2,222·9	2,239·6	2,256·4	2,273·3
54	2,290·2	2,307·2	2,324·3	2,341·4	2,358·6
55	2,375·8	2,393·1	2,410·5	2,427·9	2,445·4
56	2,463·0	2,480·6	2,498·3	2,516·1	2,533·9
57	2,551·8	2,569·7	2,587·7	2,605·8	2,623·9
58	2,642·1	2,660·3	2,678·6	2,697·0	2,715·5
59	2,734·0	2,752·5	2,771·2	2,789·9	2,808·6
60	2,827·4	2,846·3	2,865·3	2,884·3	2,903·3
61	2,922·5	2,941·7	2,960·9	2,980·2	2,999·6
62	3,019·1	3,038·6	3,058·2	3,077·8	3,097·5
63	3,117·2	3,137·1	3,157·0	3,176·9	3,196·9
64	3,217·0	3,237·1	3,257·3	3,277·6	3,297·9
65	3,318·3	3,338·8	3,359·3	3,379·9	3,400·5
66	3,421·2	3,442·0	3,462·8	3,483·7	3,504·6
67	3,525·7	3,516·7	3,567·9	3,589·1	3,610·3
68	3,631·7	3,653·1	3,674·5	3,696·1	3,717·6
69	3,739·3	3,761·0	3,782·8	3,804·6	3,826·5
70	3,848·5	3,870·5	3,892·6	3,914·7	3,936·9
71	3,959·2	3,981·5	4,003·9	4,026·4	4,048·9
72	4,071·5	4,094·2	4,116·9	4,139·6	4,162·5
73	4,185·4	4,208·4	4,231·4	4,254·5	4,277·6
74	4,300·8	4,324·1	4,347·5	4,370·9	4,394·3
75	4,417·9	4,441·5	4,465·1	4,488·8	4,512·6
76	4,536·5	4,560·4	4,584·3	4,608·4	4,632·5
77	4,656·6	4,680·8	4,705·1	4,729·5	4,753·9
78	4,778·4	4,802·9	4,827·5	4,852·2	4,876·9
79	4,901·7	4,926·5	4,951·3	4,976·4	5,001·4
80	5,026·5	5,051·7	5,076·8	5,102·2	5,127·6
81	5,153·0	5,178·5	5,204·0	5,229·6	5,255·3
82	5,281·0	5,306·8	5,332·7	5,358·6	5,384·6
83	5,410·6	5,436·7	5,462·9	5,489·1	5,515·4
84	5,541·8	5,568·2	5,594·7	5,621·2	5,647·8
85	5,674·5	5,701·2	5,728·0	5,754·9	5,781·8
86	5,808·8	5,835·9	5,863·0	5,890·1	5,917·4
87	5,944·7	5,972·0	5,999·5	6,027·0	6,054·5
88	6,082·1	6,109·8	6,137·5	6,165·3	6,193·2
89	6,221·1	6,249·1	6,277·2	6,305·3	6,333·5
90	6,361·7	6,390·0	6,418·4	6,446·8	6,475·3
91	6,503·9	6,532·5	6,561·2	6,589·9	6,618·7
92	6,647·6	6,676·5	6,705·5	6,734·6	6,763·7
93	6,792·9	6,822·2	6,851·5	6,880·8	6,910·3
94	6,939·8	6,969·3	6,999·0	7,028·7	7,058·4
95	7,088·2	7,118·1	7,148·0	7,178·0	7,208·1
96	7,238·2	7,268·4	7,298·7	7,329·0	7,359·4
97	8,389·8	7,420·3	7,450·9	7,481·5	7,512·2
98	7,543·0	7,573·8	7,604·7	7,635·6	7,666·6
99	7,769·7	7,728·8	7,760·0	7,791·3	7,822·6

THE END.